PROGRESS REPORT

PR 91565-430-4

For Month of October 1962

DEVELOPMENT OF AUXILIARY ELECTRIC POWER SUPPLY SYSTEM

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NASA Contract NAS 3-2550

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List Of Illustrations

INTRODUCTION

This report covers the work accomplished by Vickers Incorporated under NASA Contract NAS 3-2550 during the month of October, 1962. The objectives of this program are to conduct an engineering study culminating in the design of an electrical power generation system operating on hydrogen and oxygen in a space environment, and to conduct preliminary testing on critical system components.

PROGRAM SCHEDULE

The program schedule is shown in Fig. 1. The program plan for this project covering the entries of Fig. 1 was described in the progress report for July, 1962.

The schedule shown in Fig. 1 has been revised from that shown in previous program reports in order to reflect delays which have been experienced in the fabrication of experimental hardware. Parts for this program are being given top priority in the prototype shop in order to meet the revised schedule of Fig. 1.

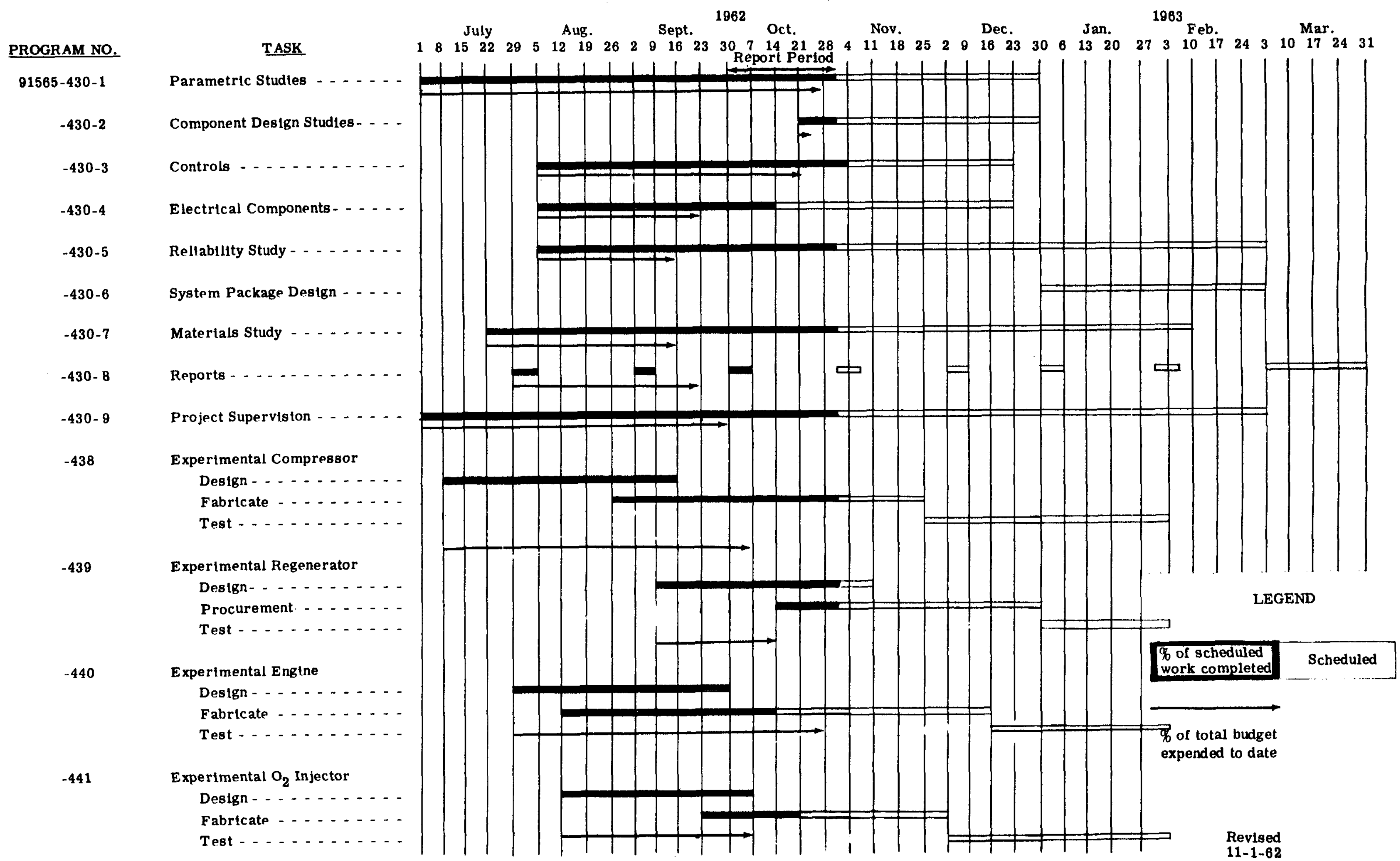
As a result of the hardware schedule slippage, testing of the experimental components will be delayed. Therefore it is recommended that the test portion of the program be extended one month, to be completed approximately February 1. This will still allow completion of testing within the original overall program schedule. At this time it appears that the revised schedule can be met and that the schedule revision will not significantly affect the overall program scope or cost.

PARAMETRIC STUDIES

In this report period the power modulation control analysis was expanded, and a hybrid system emerged as the best type of control.

Fig. 1

PROGRAM SCHEDULE NASA CONTRACT NAS 3-2550



The basic cycle analysis was also modified to include hydrogen inlet valve effects. Opening the hydrogen inlet valve before TDC has a detrimental effect on engine performance due to increased compression work. Fig. 2 shows the effect of early inlet valve opening on BSPC for three different valve durations. A minimum BSPC of .99 lb./hp-hr. or 1.47 lb./kw-hr. is possible with the Hydrox engine, using a 30^{0} duration valve. This valve duration was found to be practical from a valve dynamics standpoint.

The hybrid power modulation control system combines two modes of control; first, variable hydrogen inlet pressure up to the maximum available, then a variable phase hydrogen valve brings the engine to its maximum rated power. Engine displacement and maximum combustion pressure (and thus hydrogen supply pressure) were the principle variables in this analysis. A 30^{0} duration hydrogen valve phased for a minimum value of 2% admission was assumed. A comparison of BSPC vs. power output for various engine sizes and methods of control which would meet the power requirement is shown in Fig. 3. The lowest total propellant consumption (for the design mission) of all the power modulation arrangements analyzed is obtained by the hybrid system. An engine of 2.77 cubic-inch displacement was chosen for these three reasons:

1. When the engine is operating at a maximum combustion pressure (P_1 = 1200 psi), the BSPC is at a minimum for a 1.7 kw electrical power output.

2. The engine can operate at the required power level with a P_1 maximum = 900 psi with a very slight increase in BSPC. This is an advantage because the engine can operate directly off supercritical tankage (300 psi storage pressure) without the use of external compressors. Eliminating an external compressor will reduce system complexity and increase reliability at the expense of a small increase in propellant weight.

Fig. 2

BSPC vs. Admission

$P_1 = 1200$ psi

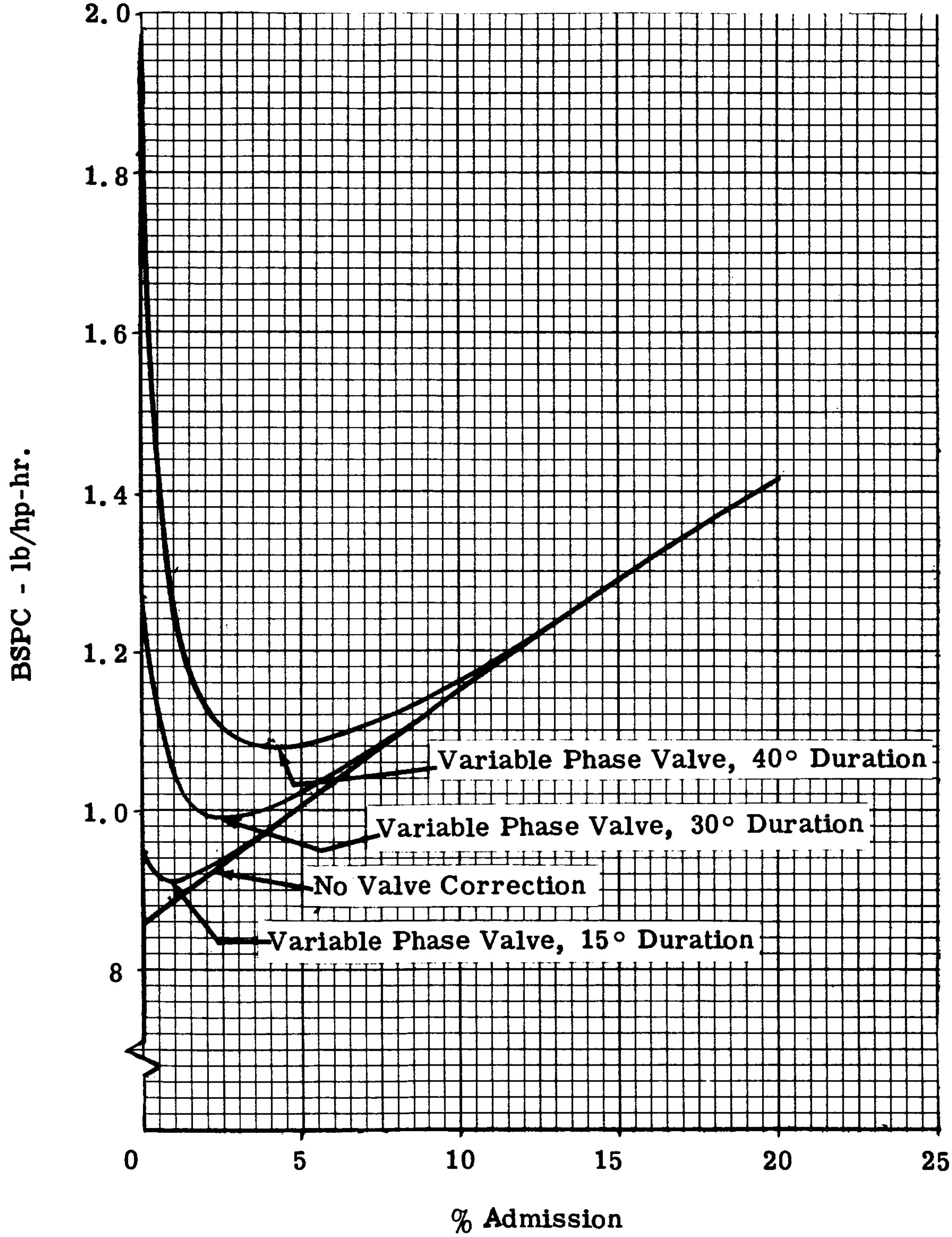


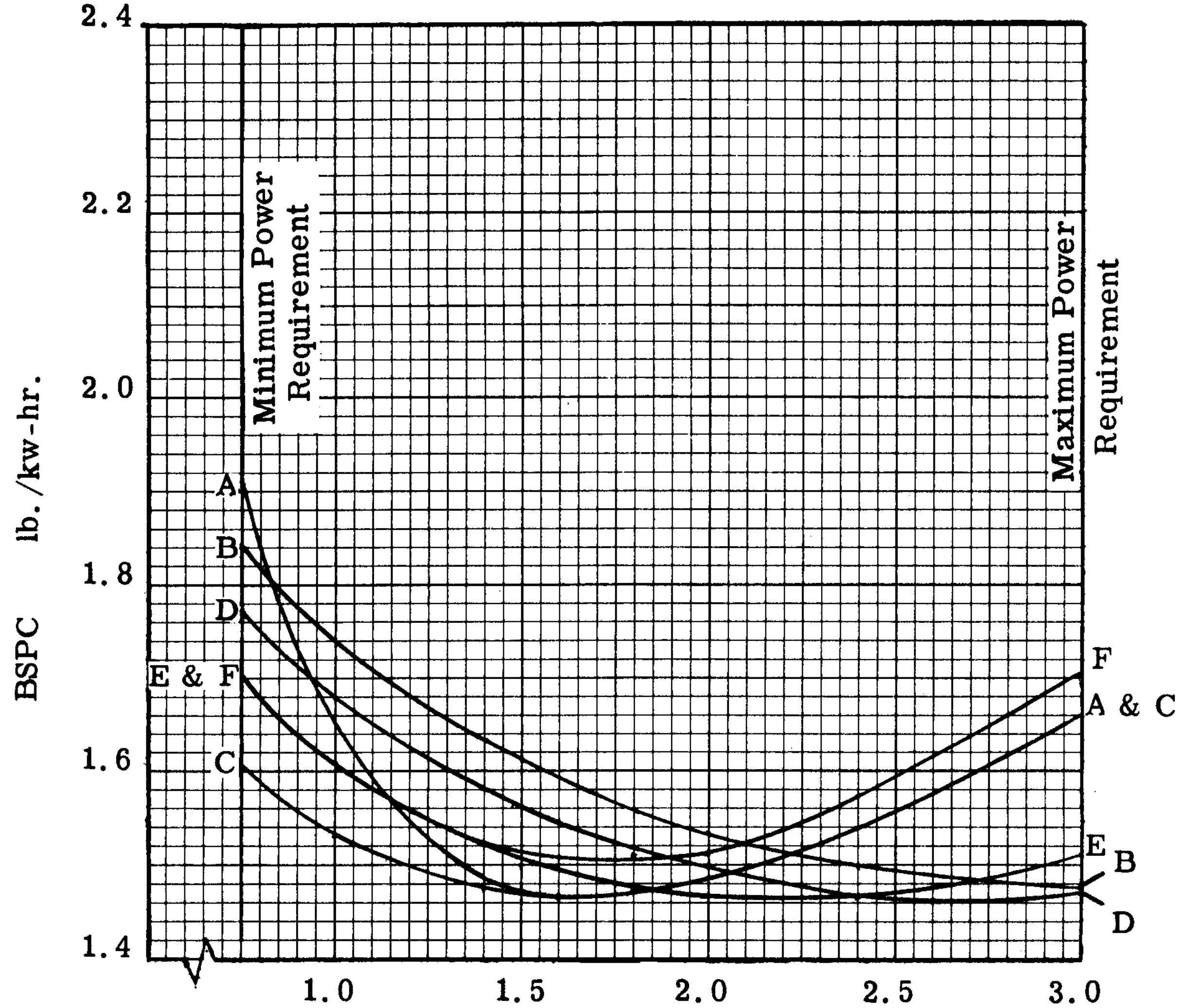
Fig. 3

BSPC vs. KW

30° Duration Hydrogen Valve

Alternator Efficiency = 90%

No Compressor Work



A. Variable phase hydrogen valve, $D = 2.08$ in.^{3}, P_1 max. = 1200 psi.

B. Variable hydrogen inlet pressure, $D = 4.2$ in.^{3}, $A = 2\%$ P_1 max. = 1200 psi.

C. Variable hydrogen inlet pressure to 1.5 kw, then variable phase hydrogen valve to full power. $D = 2.08$ in.^{3}, P_1 max. = 1200 psi.

D. Variable hydrogen inlet pressure to 2.5 kw, then variable phase hydrogen valve to full power. $D = 3.47$ in.^{3}, P_1 max. = 1200 psi.

E. Variable hydrogen inlet pressure to 2 kw, then variable phase hydrogen valve to full power. $D = 2.77$ in.^{3}, P_1 max. = 1200 psi.

F. Variable hydrogen inlet pressure to 1.4 kw, then variable phase hydrogen valve to full power. $D = 2.77$ in.^{3}, P_1 max. = 900 psi.

3. An engine of this size could be used without modification over a broadened power range (should it be required) if a moderate performance loss for power outputs of less than 0.75 kw or more than 3 kw is accepted.

Displacement was varied in an attempt to find an optimum displacement for the hybrid control system. Variable displacement implies a variable stroke engine. It is obvious from the minimum BSPC points for each displacement in Fig. 3 that if a variable stroke engine could be built it would produce minimum BSPC's over the entire power range. To date, however, there is no practical method of varying stroke in an internal combustion engine of conventional configuration.

Fig. 3 also shows the BSPC comparison between operation at P_1 = 1200 psi (curve E) and P_1 = 900 psi (curve F). This comparison shows that for a 2.77 cubic-inch engine the curves are the same below 1.45 kw, which is the throttled inlet range below P_1 = 900 psi. For power outputs greater than 1.45 kw, the BSPC is shown in curve F. Since the average power for the design mission is about 1.7 kw, it can be seen from curve F that the engine would be operating at near minimum BSPC for most of the mission. The additional fuel for operating the total mission at P_1 = 900 psi instead of P_1 = 1200 psi would be less than 10 lbs. for a 2.77 cubic-inch engine, (see Table I). This is a small penalty for eliminating a hydrogen compressor. These operating pressures are possible with supercritical storage, which requires tank pressures of 300 psi for hydrogen and 1000 psi for oxygen. Conventional tankage with high reliability could be utilized.

A further increase in storage pressure to 400 psi for hydrogen and 1300 psi for oxygen (which would be necessary for 1200 psi combustion pressure) would save 10 pounds of fuel and add 100 pounds or more of tank weight, resulting in a higher total power system weight. If P_1 maximum is lowered much below 900 psi, the total system weight will again increase, since the BSPC would increase, and additional fuel would be required without lowering tankage weight. Therefore, P_1 maximum of approximately 900 psi appears to be near optimum.

Table I lists the total propellant that would be required by each of the engine configurations in Fig. 3 to complete the design mission. The BSPC's shown in Fig. 3 have been increased by 10% in Table I to cover compressor work on residual propellants trapped in the supercritical storage tanks. The propellant consumption is divided into two parts, 0-100 hrs. and 100-350 hrs., since "boil-off" propellant may be available during part of the first 100 hrs.

TABLE I

Time	Engine Configurations In Fig.					
	A	B	C	D	E	F
0-100 hours	243	266	242	257	247	248
100-350 hours	674	724	672	703	682	690
Total	917	990	914	960	929	938

Beech Aircraft has been contacted regarding tankage methods. Their opinion is that only supercritical tankage should be considered for this power system at present. Beech Aircraft is starting a study on subcritical expulsion, which could change their opinion if positive results are obtained.

COMPONENT DESIGN STUDIES

The following items will be investigated in the component design studies:

1. Flight engine configuration
2. Flight engine cooling system
3. Oxygen injector location and orientation
4. Heat rejection control
5. Zero-g lubrication system
6. Dynamic balancing of entire system
7. Vibration isolation

Immediate emphasis will be on items 5 and 6. These considerations do not depend strongly on the results of performance tests, but are of vital importance to overall system design. Since all flightweight system design hinges on item 1, effort has been concentrated in this direction during this report period.

Flight Engine Configuration

The results of hydrogen valve dynamic studies are shown in Figs. 4 and 5. A first look at gas flow characteristics is shown in Fig. 6. These studies support the feasibility of a poppet valve of approximately 0.25" diameter and 0.035" lift for a 30° duration. Poppet seating force at these loads may be high enough with a conventional valve spring to require desmodromic valve closing with a spring on the other side of a rocker arm. The feasibility of driving both hydrogen and oxygen valves from cams mounted directly on the crankshaft through similar torsion arrangements is also under consideration.

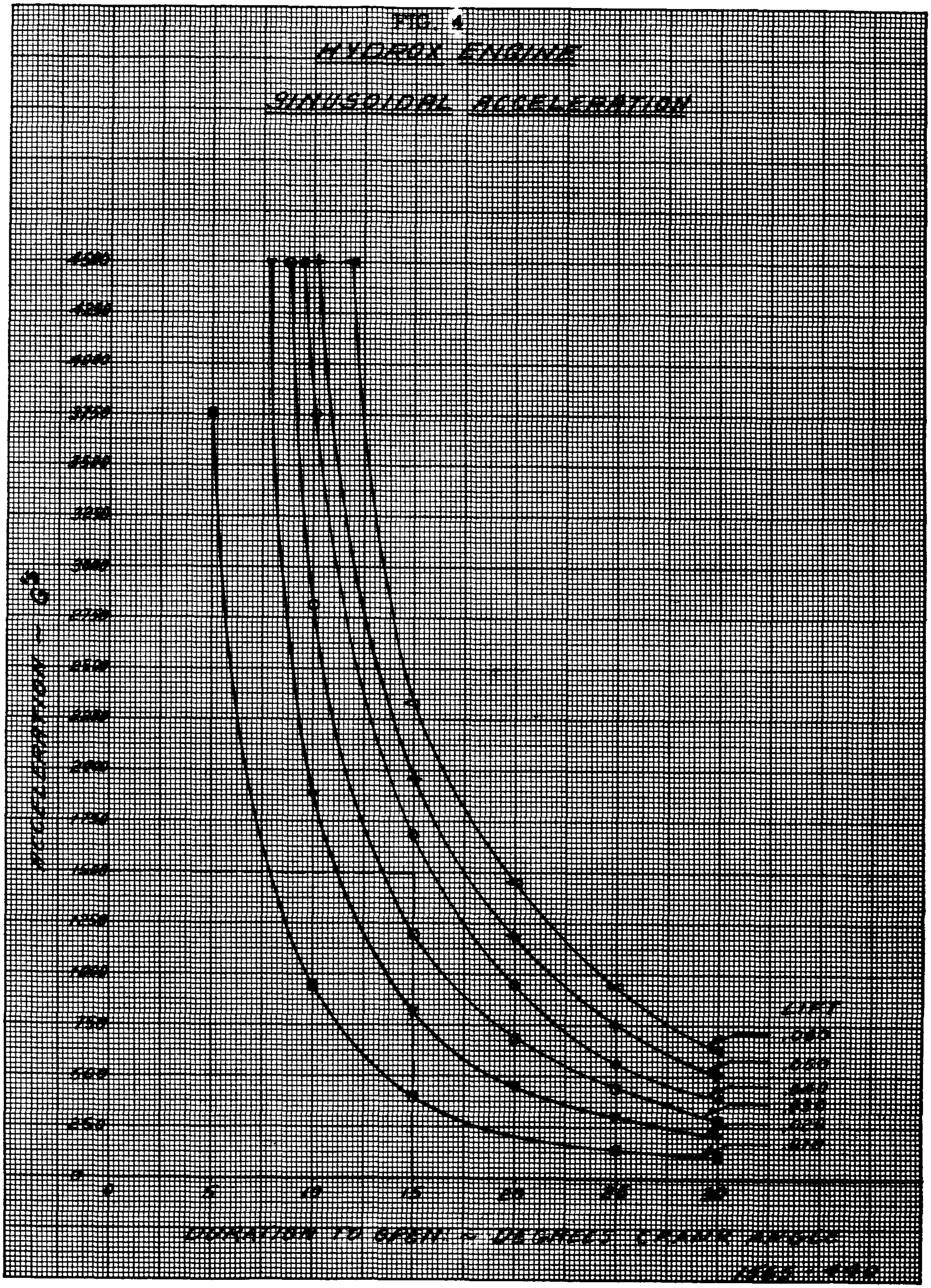


FIG. 4
HYDROX ENGINE
SINUSOIDAL ACCELERATION
ACCELERATION ~ G's
DURATION TO OPEN ~ DEGREES CRANK ANGLE
LIFT
0
5
10
15
20
25
30

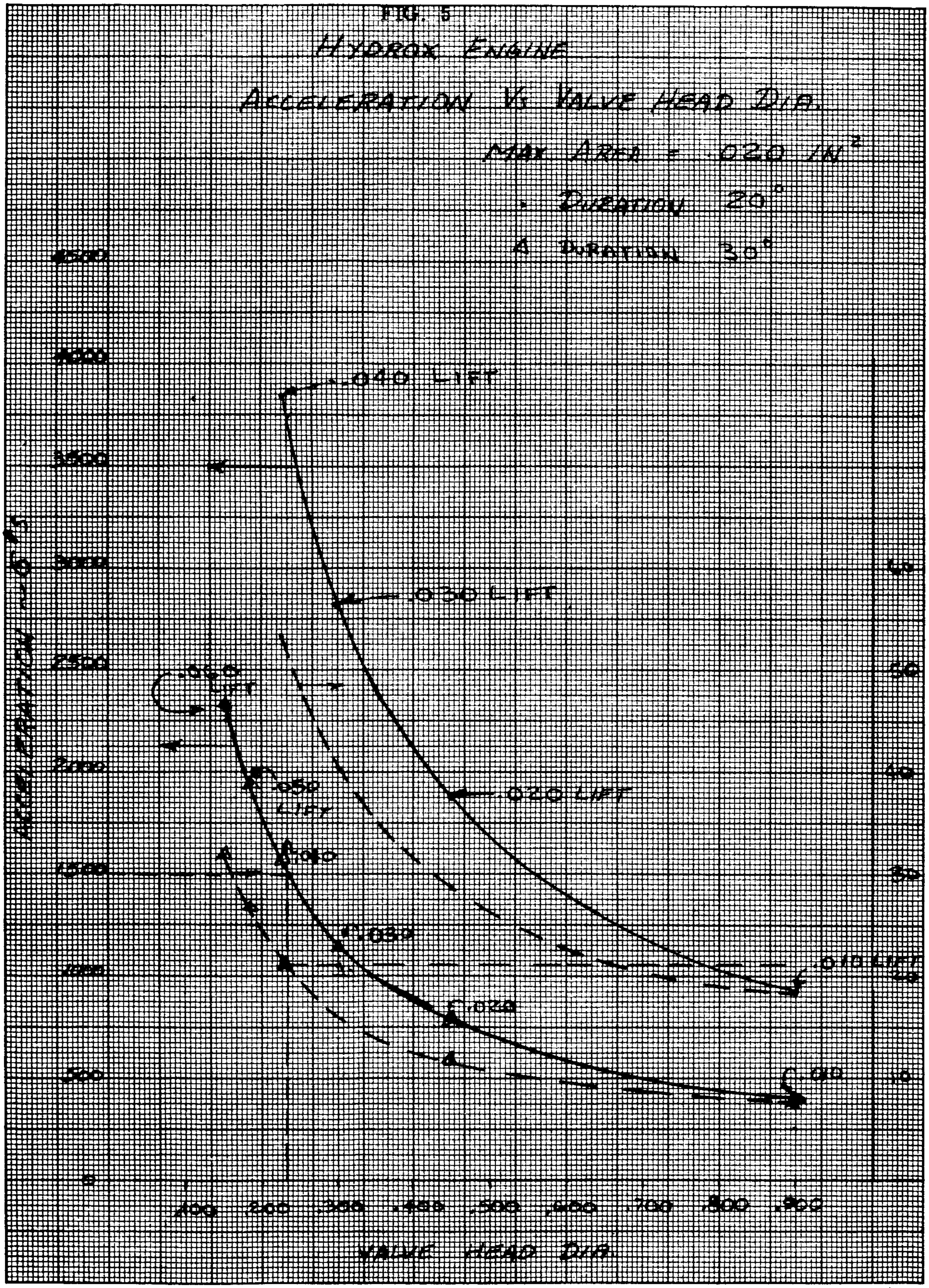


FIG. 5
HYDROX ENGINE
ACCELERATION VS VALVE HEAD DIA.
MAX AREA = .020 IN²
• DURATION 20°
▲ DURATION 30°
.040 LIFT
.030 LIFT
.060 LIFT
.050 LIFT
.020 LIFT
.010 LIFT
.040
.030
.020
.010
ACCELERATION ~ G'S
SPRING FORCE ~ LBS
VALVE HEAD DIA.
4500
4000
3500
3000
2500
2000
1500
1000
500
0
.100
.200
.300
.400
.500
.600
.700
.800
.900
60
50
40
30
20
10

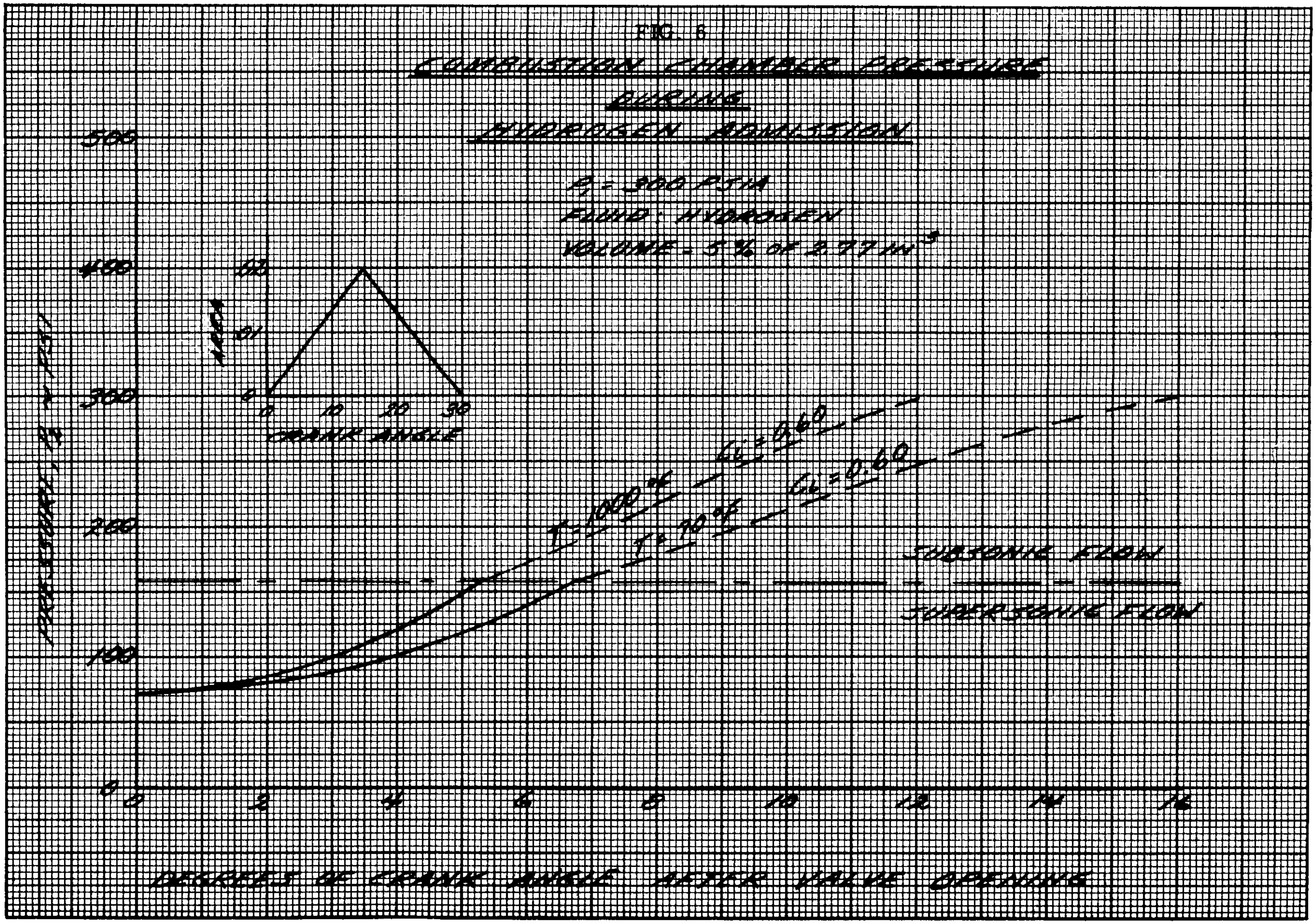

COMBUSTION CHAMBER PRESSURE
DURING
HYDROGEN ADMISSION
P1 = 300 PSIA
FLUID: HYDROGEN
VOLUME = 5% OF 2.77 IN3
PRESSURE, P, ~ PSIA
500
400
300
200
100
0
AREA
.02
.01
0
0
10
20
30
CRANK ANGLE
T = 1000 °F
T = 70 °F
CL = 0.60
CL = 0.60
SUBSONIC FLOW
SUPERSONIC FLOW
0
2
4
6
8
10
12
14
16
DEGREES OF CRANK ANGLE AFTER VALVE OPENING

FIG. 6

In Fig. 4, maximum acceleration in g's is plotted against crank angle degrees (from opening to maximum lift) for various lifts. Sinusoidal acceleration is assumed. 1500 g's is the current design practice for racing engines. In Fig. 5, acceleration and spring force are shown as functions of valve head diameter for various lifts and for two durations (20° and 30° total duration, corresponding to 10° and 15° opening duration in Fig. 4). A maximum area of 0.020 in.^{2} is assumed. The spring force of 22 lbs. may seem quite high, but is based on current valve component weights and is therefore conservative.

Hydrogen flow into a cylinder with piston at TDC and with the maximum area of 0.020 in.^{2} used in the dynamics analysis is shown in Fig. 6. Both regenerated (1000°F) and unregenerated (70°F) are shown. A linear area change vs. crank angle as shown in the insert was used. It can be seen that cylinder filling with a 30° valve duration will not be a problem, and that increased compression work with the 2% to 5% phasing under consideration for the hybrid cycle (corresponding to crank angles between 14° and 23° ATDC) will be negligible. Further studies using an actual cam profile and considering C (flow coefficient) as a function of lift will be considered.

CONTROLS

The controls report for this period will cover the following:

A. Engine control configuration analysis
B. Evaluation of propellant source control
C. Evaluation of control components
D. Evaluation of mode of control

A. Engine Control Configuration Analysis

Control methods for the hybrid power modulation techniques have been investigated. This hybrid system requires that means of control previously considered in a separate manner must now be combined in actual system operation. The effect of this change in terms of system complexity is to add a hydrogen throttle valve in series with the variable phased valve (see Fig. 7). A slightly more complex manner of scheduling valve areas and phasing mechanism position is required, but if the proposed mechanical approach is feasible the resulting system adds very little cost in terms of reliability or added components.

A simplified mechanical schematic (see Fig. 8) is included to show the concept of mechanization. Power setting can be thought of as being proportional to electromechanical actuator position. Flat areas on cams are utilized to phase out the modulating effects of the valve or phasing cam.

Evaluation of the inertial effect of heavier generator designs indicate that the system will have a moment of inertia 20 times that originally calculated, and 5 times the minimum design value indicated in Fig. 6 and 7 of the October Progress Report. The error is accord-

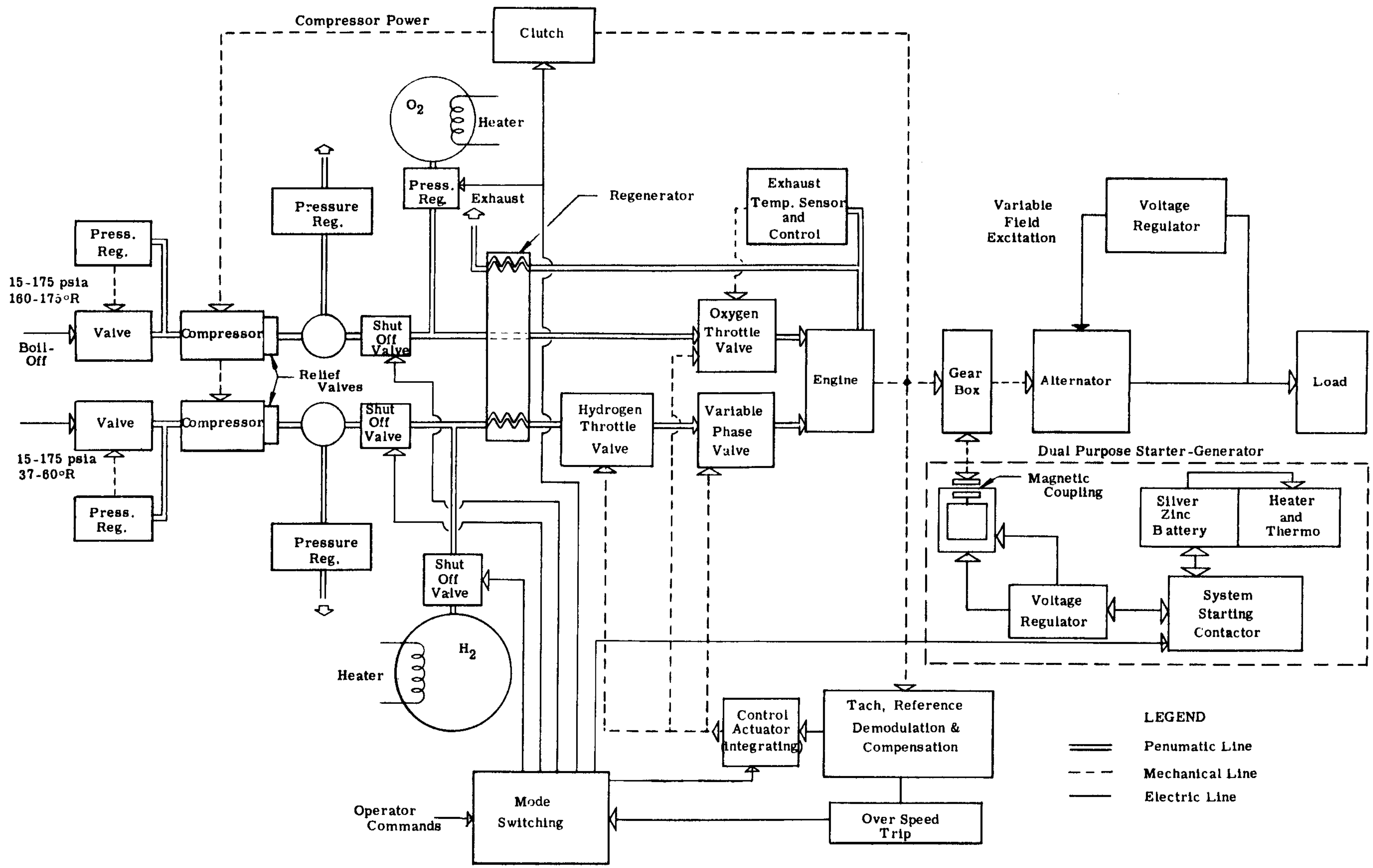


Fig. 7 - Control Schematic Block Diagram

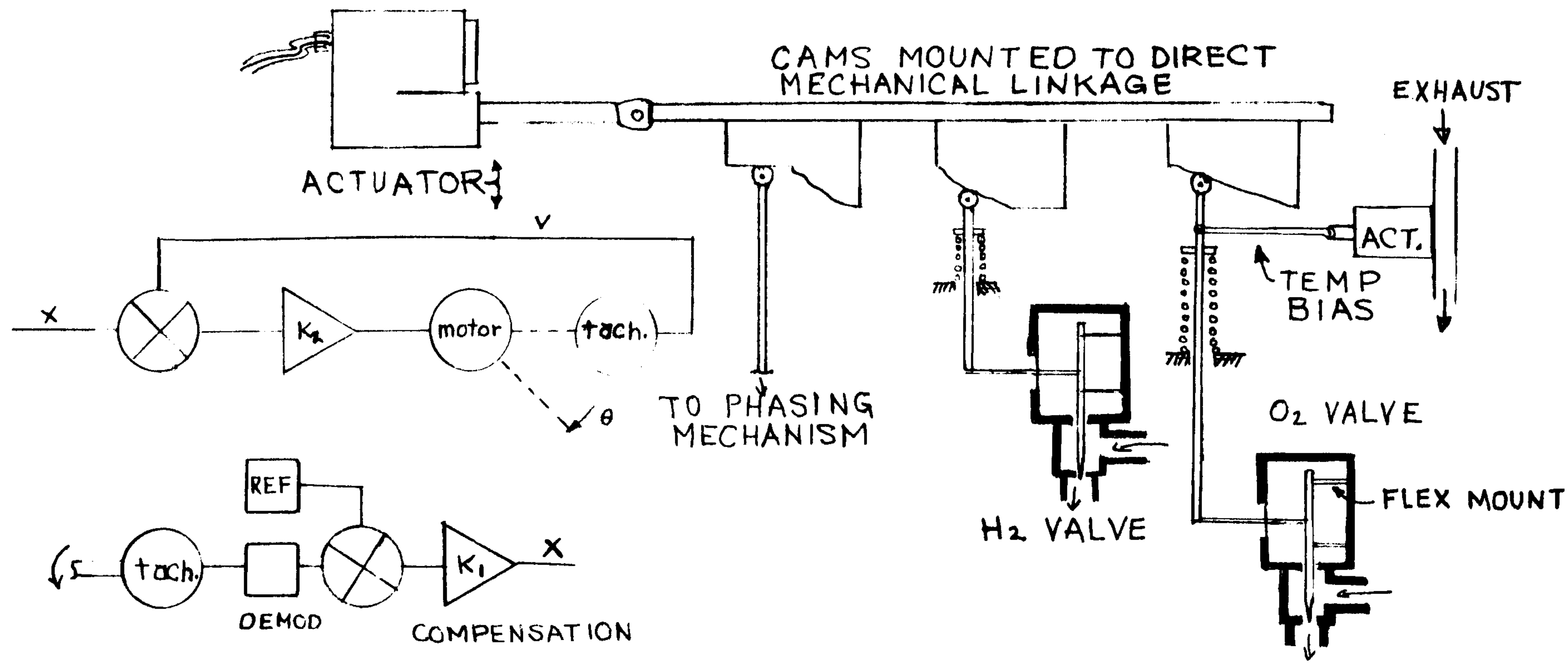


Fig. 8 - Control System Schematic

ingly reduced for a given system gain, or the system can be operated at substantially lower gain, eliminating the requirement for load sensing and reducing the actuator and valve response requirements.

B. Evaluation of Propellant Source Control

Previously, oxygen and hydrogen stored in the tanks was compressed before admission to the engine. An evaluation of tankage pressures and of compressor requirements necessary to utilize oxygen and hydrogen at the previously designated pressures indicated an overall advantage in engine operation at supercritical tankage pressures. This change is shown in Fig. 7. The oxygen is now heated slightly in the tank to avoid very low temperature at the throttling valve and at the engine. Provisions have been added to allow oxygen from the tank to supplement the boil-off supply in the event it is not equal to the demand. The compressor may be de-clutched allowing the system to operate at a more efficient level when boil-off is not used.

Careful consideration was given to the control of the compressor. Previously, a closed loop control around the compressor was proposed. Although this system is feasible, the following problems become obvious:

1. If the compressor is sized for pumping the maximum fuel rate required when the inlet pressure is 15 psia, then in order to regulate flow rates down to the minimum (about 50% of maximum) the inlet pressure must be reduced to about 7.5 psia. For constant exhaust pressure, the pressure ratio will increase as the inlet pressure is throttled. The higher exhaust temperatures which result are undesirable.

2. If the inlet pressures are allowed to get too high, the work load on the engine goes up accordingly and this over-load affects sizing of the engine. A relief valve would be required to avoid this overpressure.

3. Closing the loop around the compressor using very simple and easily mechanized techniques does not give desirable results, primarily due to the valve gain variation upstream of the compressor. This variation in gain is due to the wide range of inlet pressures from boil-offs as well as the change in flow coefficient of an unchoked flow orifice. On Fig. 9, runs A, B, C, D, E, F, and G are analog traces of the results of this type of system in terms of throttled compressor inlet pressure. If gain compensation is added to correct for this poor response, the results could be represented on the same traces, within the range of runs C to D. Adding gain compensation would add complexity to the system mechanization.

The approach proposed at present is based on the following ground rule:

<u>While operating on boil-off, a slight degradation in S.P.C. at less than maximum power settings can be tolerated</u>.

The proposed approach is to regulate compressor inlet and exhaust pressures to constant values, retaining the feature to avoid high inlet pressures at any time. Compressor flow, work, and pressure ratio are now essentially constant. The mismatch between compressor delivery and engine demand flow rates is to be bypassed, either overboard or to some useful purpose. The work in compressing the bypassed flow is lost.

Fig. 9 - CLOSED LOOP COMPRESSOR CONTROL FOR HYDROGEN COMPRESSOR

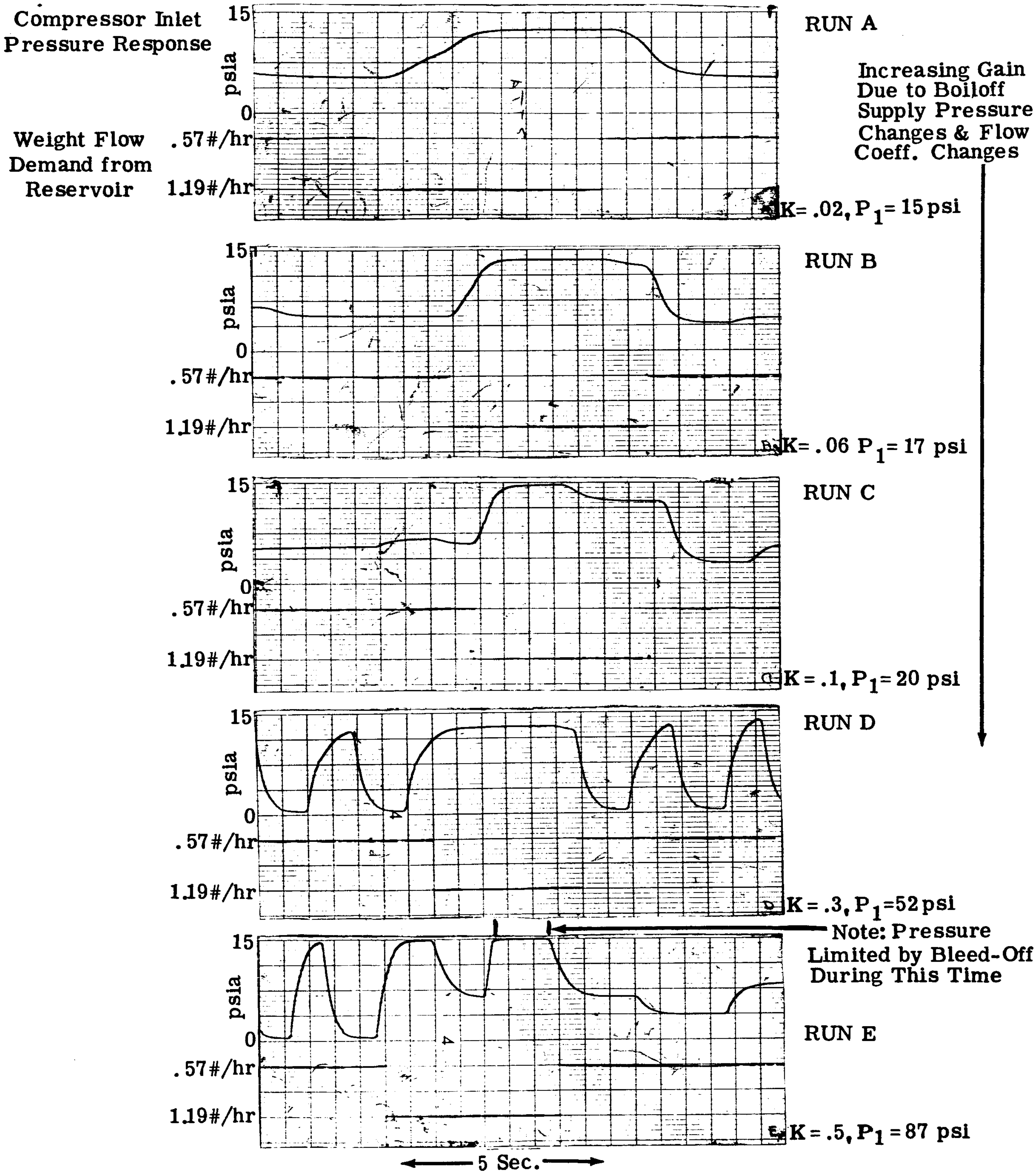


Fig. 9 - Cont.
CLOSED LOOP COMPRESSOR CONTROL
FOR HYDROGEN COMPRESSOR

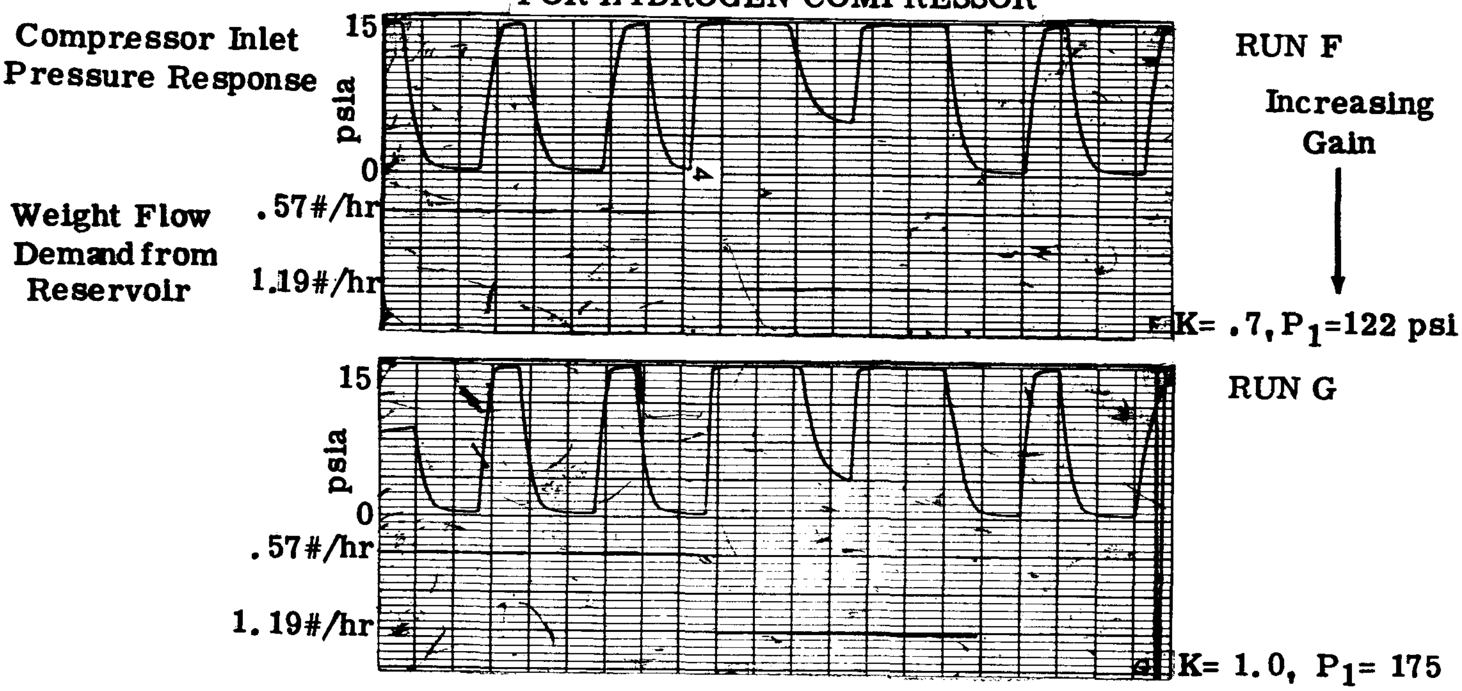
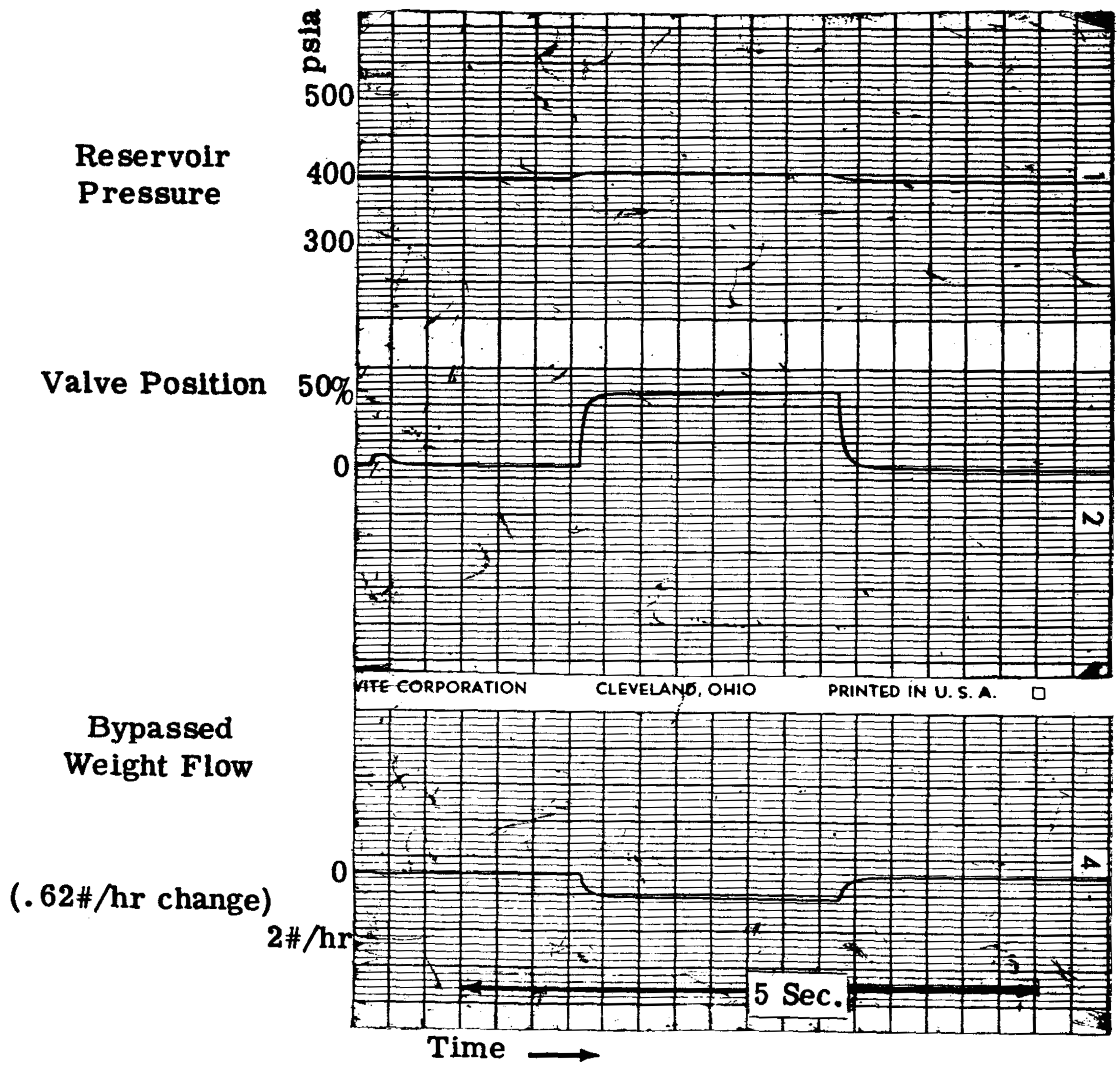


Fig. 10 - TYPICAL PRESSURE REGULATOR RESPONSE
HYDROGEN RESERVOIR PRESSURE REGULATOR



The only controls necessary consist of the compressor inlet pressure regulator and the reservoir pressure regulating relief valve which bypasses the flow necessary to maintain a constant pressure. Both of these units perform functions similar to many regulators already designed. The response of a typical regulator is shown in Fig. 10. Since the loop is not closed, problems associated with closing it are eliminated.

C. Evaluation of Control Components

Valves:

Probably one of the biggest problems in component selection or design is to find an adequate valve to operate at the cryogenic temperatures encountered in regulating the compressor supply and in supplying gases to the system from the supercritical tanks. Although a number of potential suppliers for these and other valves throughout the system have been contacted, very little has been accomplished in locating actual hardware to accomplish the quite specialized tasks. Requirements for the valve are fairly well established at this time, which will expedite future contact.

Actuator:

A single actuator to move all power modulating components has evolved with the hybrid control concept. Various configurations under consideration for this actuator involve:

1. Magnetic clutches
2. Two-phase motors
3. Torque motors
4. Hot gas or hydraulic power

Of these concepts, magnetic clutches or 2-phase motors are the most applicable, and the 2-phase motor is the more standard and tested method. Further evaluation will follow in this area.

Control System:

Other system components are: a-c tachometers, amplifiers, modulators and demodulators, power supply and reference signals supply. A preliminary evaluation of the functions performed by these components indicate that a working system can be obtained through the use of standard type components; however, a system fully optimized with respect to reliability, weight, simplicity, packaging, etc., would be beyond the scope of this study.

D. Evaluation of Mode of Control

Components and signals necessary to select and maintain the various modes of control to which the engine will be subjected have been added to the control block diagram, (Fig. 7). The accomplishment of these functions is performed through the logical switching of various components at the command of sensors within the system or by the operator. Table II has been prepared to outline the state of all components involved versus the mode of control.

TABLE II

MODE	Oxygen Tank Pressure Regulator	Hydrogen Tank Pressure Regulator	Compressor Shutoff Valve, Oxygen	Compressor Shutoff Valve, Hydrogen	Compressor Clutch	Starting Contactor	Power Supplies Contactor	Control Override					
STARTING	C	C	O	O	C	C	C	C					
OPERATE BOILOFF	C	C	O	O	C	O	C	O					
OPERATE TANKAGE	O	O	C	C	O	O	C	O					
OVERSPEED, OR FAILSAFE	C	C	C	C	O	O	O	O					
WARMUP	C	C	C	C	O	O	C	C					

O - Open

C - Closed

ELECTRICAL COMPONENTS

Vendors contacted to date have been General Electric, Westinghouse, American Mark, and Curtiss Wright. Response in proposal form is forthcoming but has not been received during this report period. Vickers effort will resume upon receipt of vendor proposals.

MATERIALS

Materials under consideration which meet the NASA high temperature requirements are given in the October progress report. Further materials investigation is proceeding.

OXYGEN INJECTOR

Fabrication of the oxygen injector is approximately 33% complete.

HYDROGEN COMPRESSOR

Fabrication of the hydrogen compressor is now approximately 75% complete.

EXPERIMENTAL ENGINE

Fabrication of the experimental engine is approximately 25% complete. The test set-up will be essentially the same as that used during the previous ASD contract except that an American Instrument Company Recording High Speed Engine Indicator Catalog No. 5-1711 will be used with a balanced diaphragm pick up to obtain indicator diagrams. This recording device was developed by MIT and is described briefly in pages 211 thru 213 of "The Internal Combustion Engine" (Second Edition) by Taylor and Taylor. This recorder provides balanced diaphragm accuracy without having to resort to the laborous data gathering and data reduction processes associated with the point by point techniques. In addition, the recorder requires a minimum of calibration. The recorder is now being procured.

Problem Areas

No problem areas beyond those originally anticipated for this program have been encountered during this report period. New technical problems are not expected until component testing is under way.

Planned Future Work

For the forthcoming report period it is planned to continue work in all of the areas of the program indicated on the schedule in Fig. 1. Parametric studies, controls analysis, selection of electrical components, reliability studies, and material studies will be continued. Fabrication of all experimental components will continue.

Component design studies will investigate flight engine dynamics, balancing, and lubrication systems.

RELIABILITY STUDIES

The principal reliability effort in October was a continuing study of the reliability of various low power reciprocating engines for the purpose of establishing a valid reliability estimate for the Hydrox engine. This estimate will be of prime importance in predicting system reliability. Several well known engine manufacturers were contacted and useful replies were received from the Briggs and Stratton, and Wisconsin Engine Corps. The information and failure rate data received was taken from engines using hydrocarbon fuels, and therefore will apply only in part. For example, spark plug fouling and carbon and lead deposits must be discounted as possible failure modes in a Hydrox engine. Conversely, the Hydrox engine may demonstrate certain problem areas considered unlikely in a gasoline engine during a 14-day mission, such as ring failure due to high temperature cylinder walls. Other engine manufacturers contacted to date with a reply forthcoming are: McCulloch, Clinton, Continental and Lycoming.

Some preliminary component reliabilities were estimated during October. Since flight system component designs are not yet finalized, certain critical characteristics needed for estimating failure rates are not known. Some design areas are now becoming more definitive, and initial reliability estimates may be made during November.

The tentative program for reliability effort remains as follows:

1. Estimate MTBF for significant components.
2. Conduct failure mode analysis.
3. Determine redundancy requirements.
4. Recommend fail-safe techniques.

REGENERATOR

A. Procurement

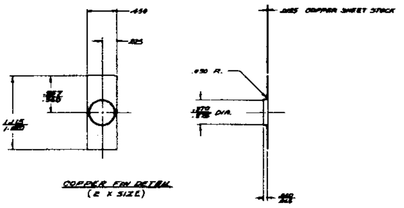
A regenerator specification was written (See Appendix A) and a regenerator design layout (Figs. 11 - 16) was drawn. Both the specification and layout were submitted to vendors with instructions to bid on the fabrication of: (1) a regenerator of their own design which meets the requirements of the specification; (2) the regenerator shown in the layout; and (3) any modification of the regenerator shown in the layout which meets the requirements of the specification and also reduces fabrication time and costs. The possibility of building the regenerator in-house was also considered, but this approach was rejected because of the lack of facilities to perform all of the fabrication operations and the desire to place complete responsibility upon a single vendor.

The reasons for the above approach are as follows:

1. The specification allows vendors with both analytical and fabrication capabilities to demonstrate their ingenuity.

2. The design layout makes it possible to solicit bids from vendors lacking analytical ability.

3. The knowledge obtained through the design study, and consultation with fabricators and processors involved in the preparation of the design layout, places Vickers in a better position to evaluate vendor designs.



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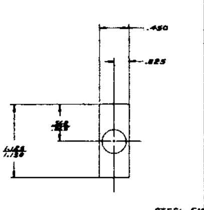
(2 x SIZE)



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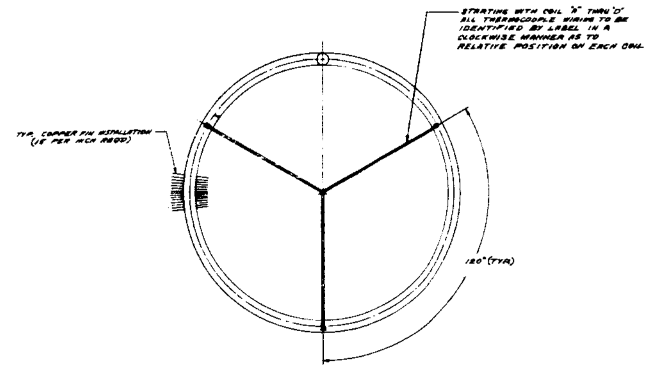
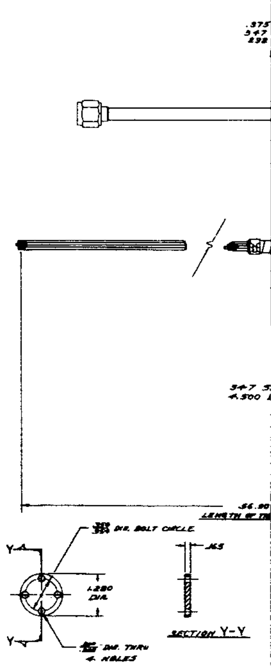
(2 x SI



TYPICAL THERMOCOUPLE WIRE ATTACHMENT

TYPICAL OF COILS "A" "B" "C" & "D"



2.- NICKEL OR SILVER BRAZE COPPER FINS (VENDORS OPTION)

1.- ALL WELDS TO BE OF THE HELI-ARC TYPE

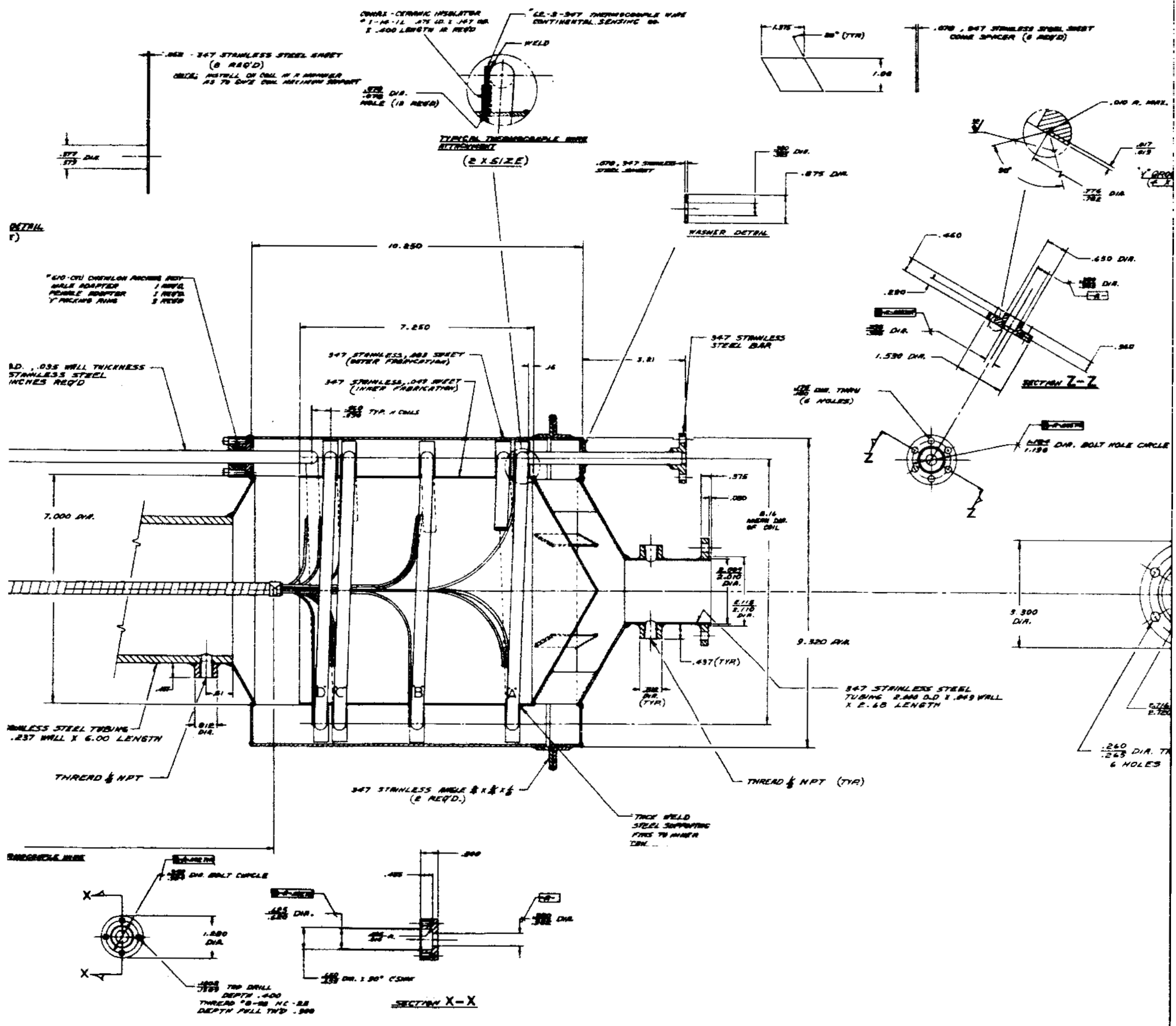
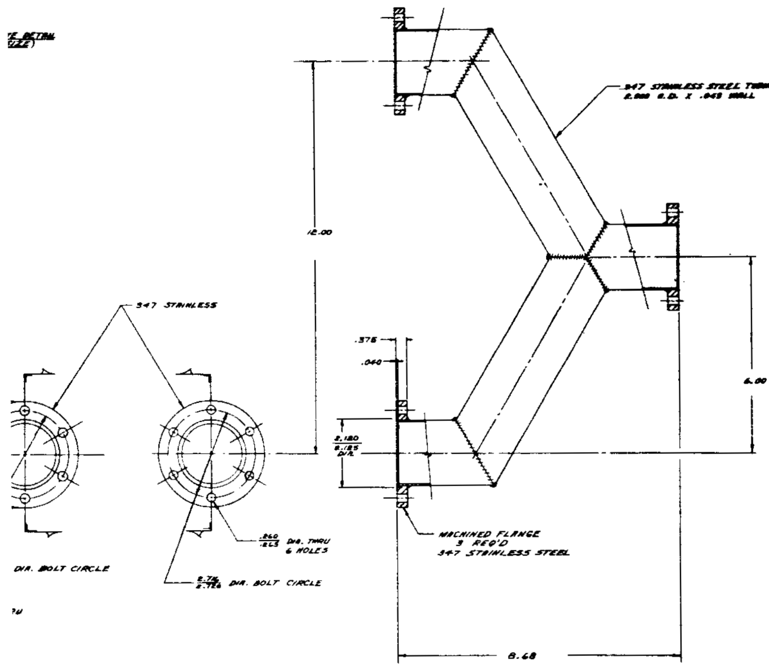


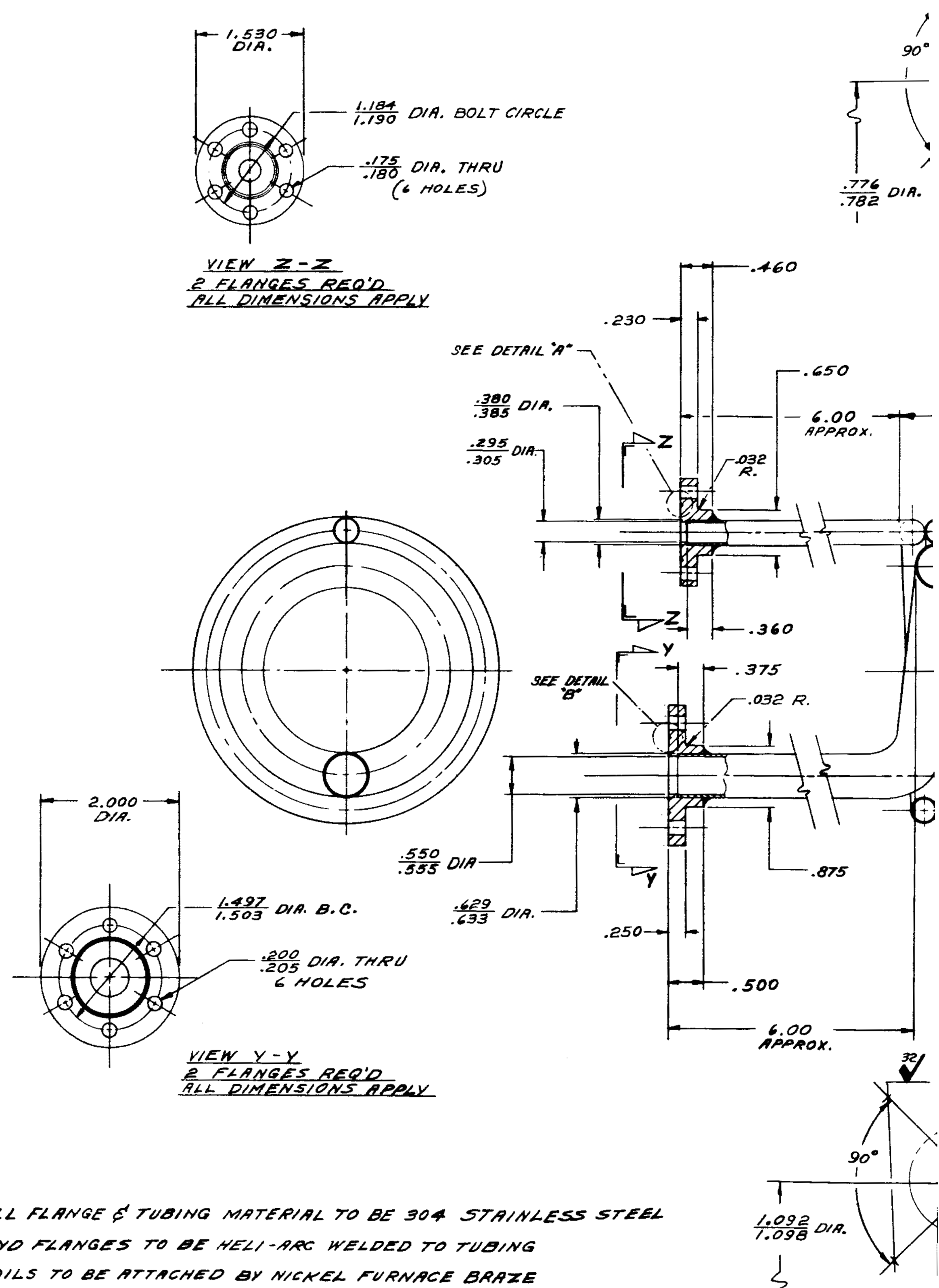
Fig. 11 - Exhaust - Hydrogen Regenerator

2



347 STAINLESS STEEL TUBING
2.000 O.D. X .049 WALL
12.00
347 STAINLESS
.376
.040
6.00
MACHINED FLANGE
3 REQ'D
347 STAINLESS STEEL
.260 / .263 DIA. THRU
6 HOLES
DIA. BOLT CIRCLE
DIA. BOLT CIRCLE
8.68

3



5 ALL FLANGE & TUBING MATERIAL TO BE 304 STAINLESS STEEL

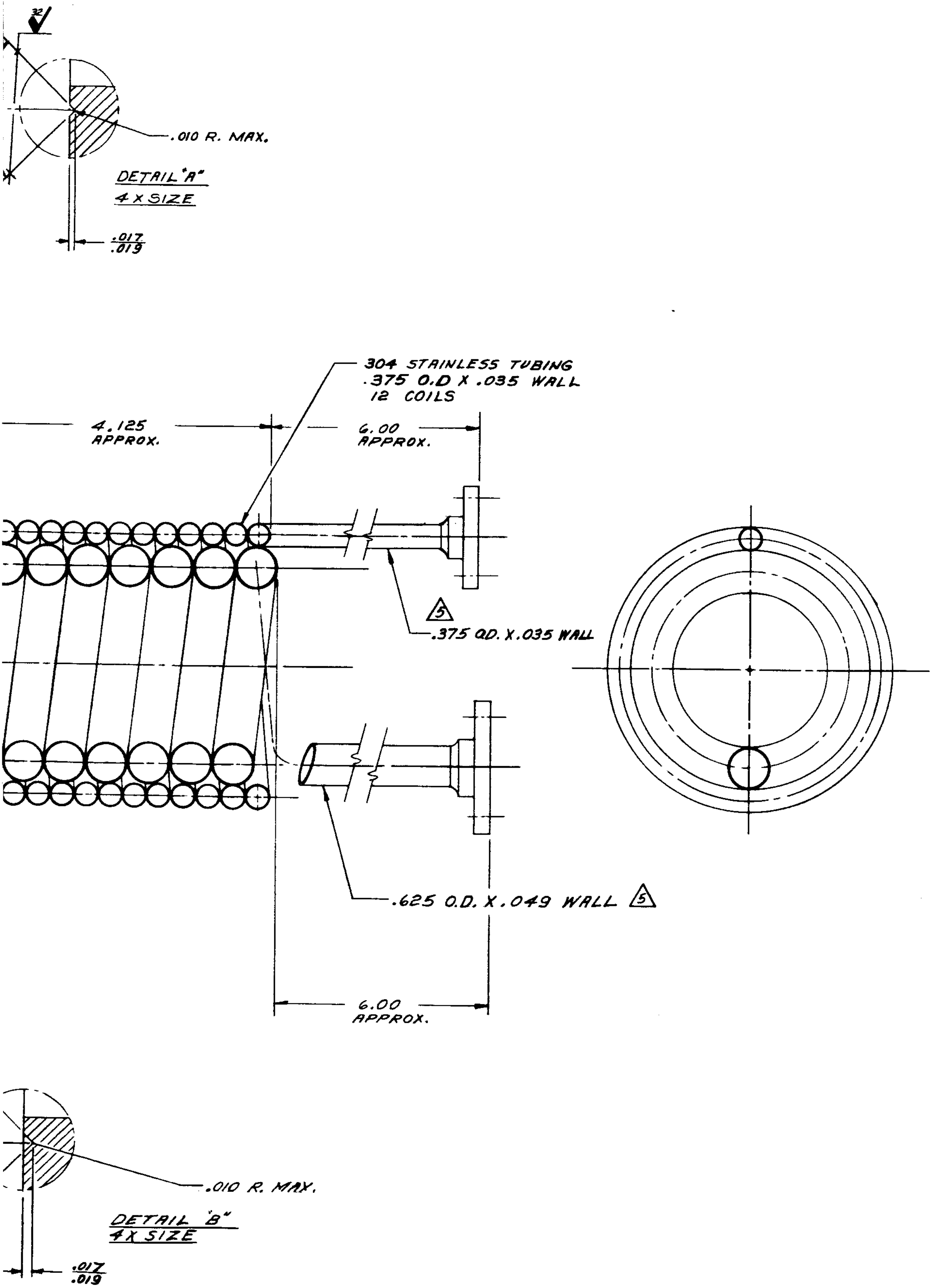
4 END FLANGES TO BE HELI-ARC WELDED TO TUBING

3. COILS TO BE ATTACHED BY NICKEL FURNACE BRAZE

2. UNITED AIRCRAFT METAL "O" RING 1.125 O.D. X 1/32 C.S. U-6105-01125-PFA-2 REQ'D

1. UNITED AIRCRAFT METAL "O" RING .875 O.D. X 1/32 C.S. U-6105-00875-PFA-2 REQ'D

Fig. 12 - Hydrogen-Oxygen H



eat Exchanger

2

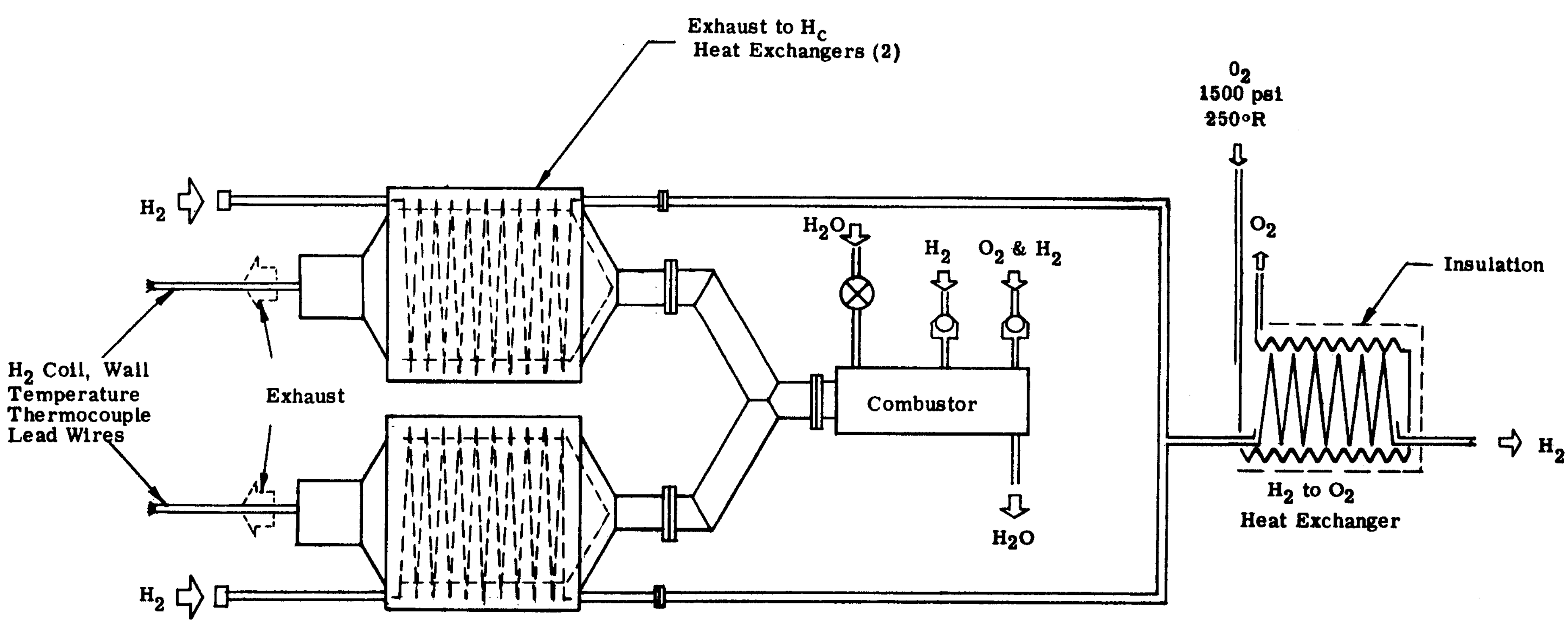


Fig. 13 - Schematic Heat Exchange Set-Up

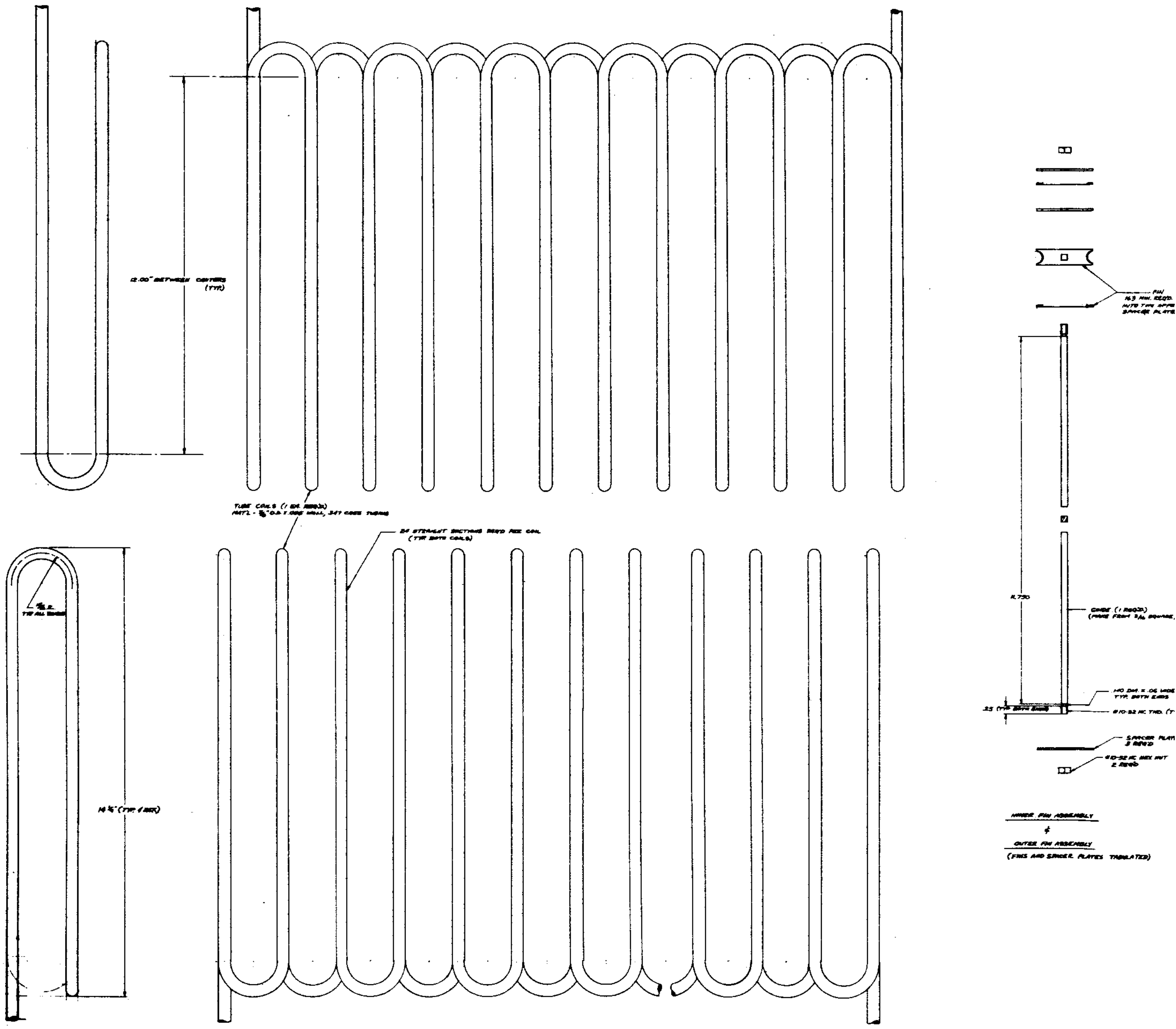
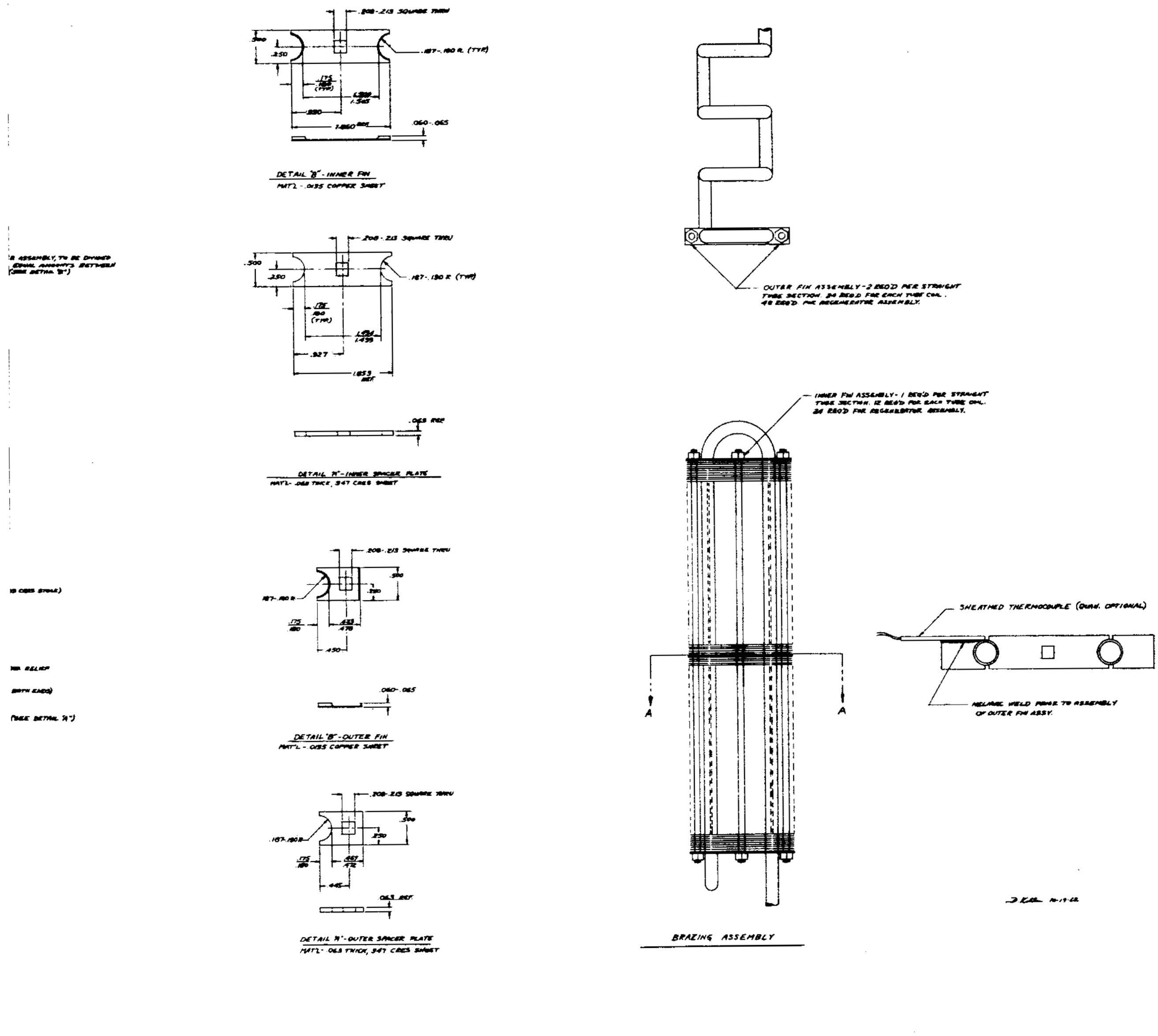


Fig. 14 - Regenerato
Rectangula



· **Design Study**

· **Configuration**

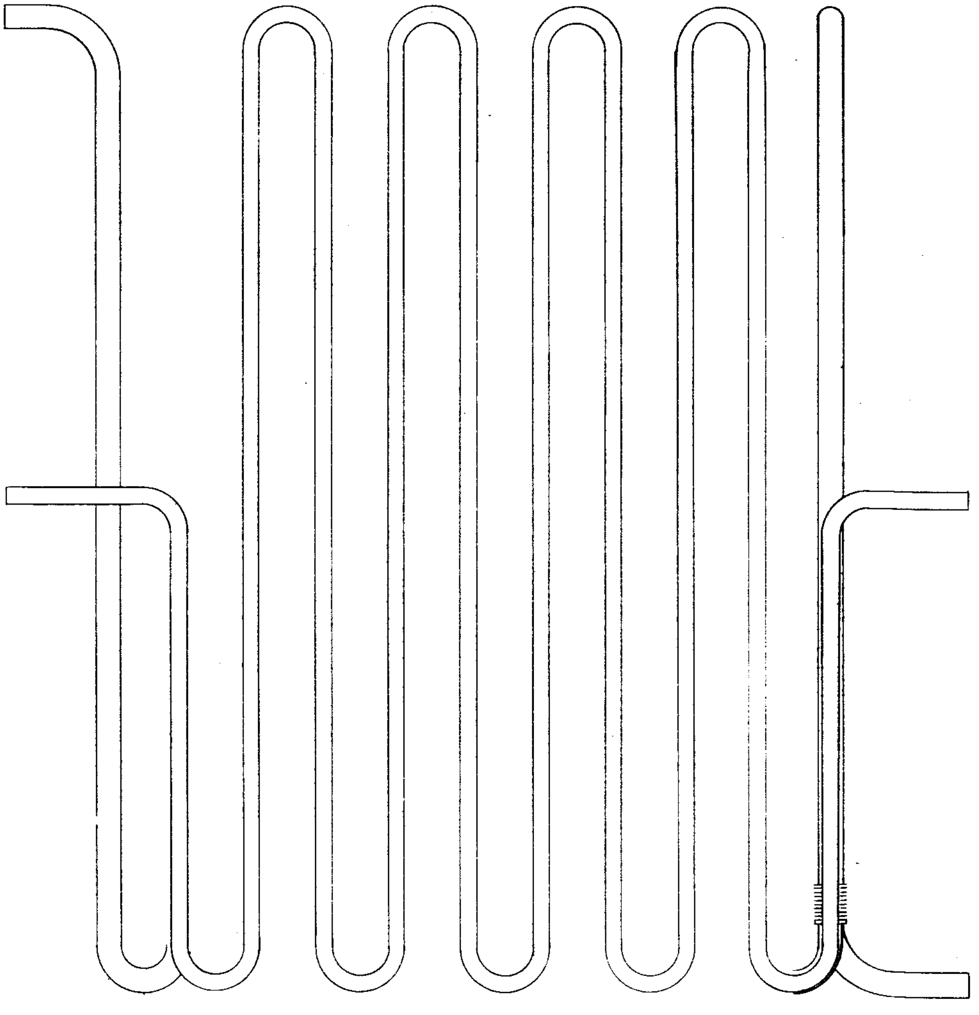
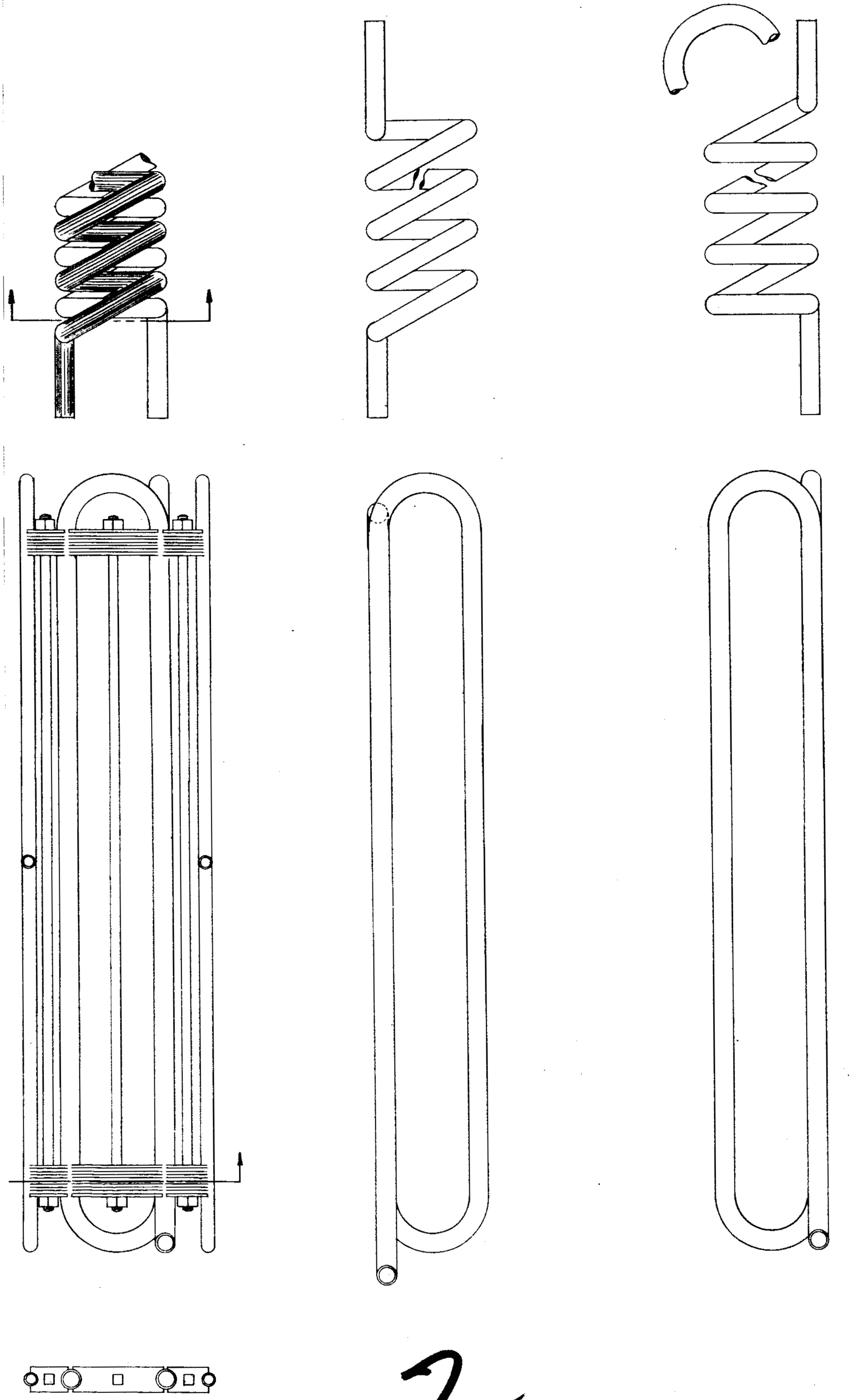
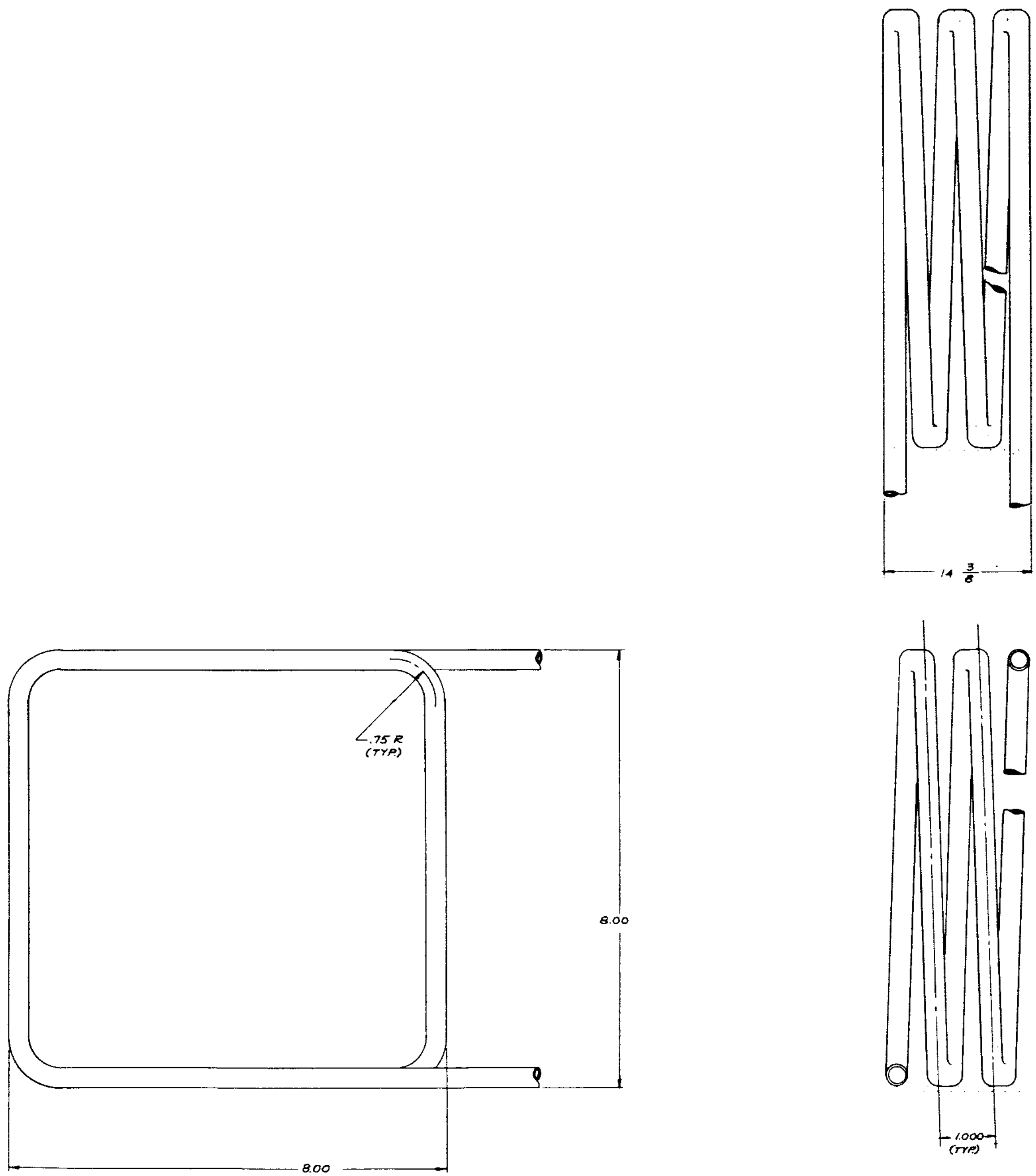


Fig. 15 - Reg
Re



2

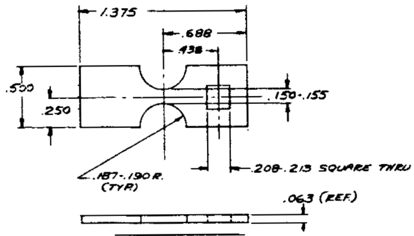
enerator Design Study
tangular Configuration



COILED TUBING (2 REQ'D)

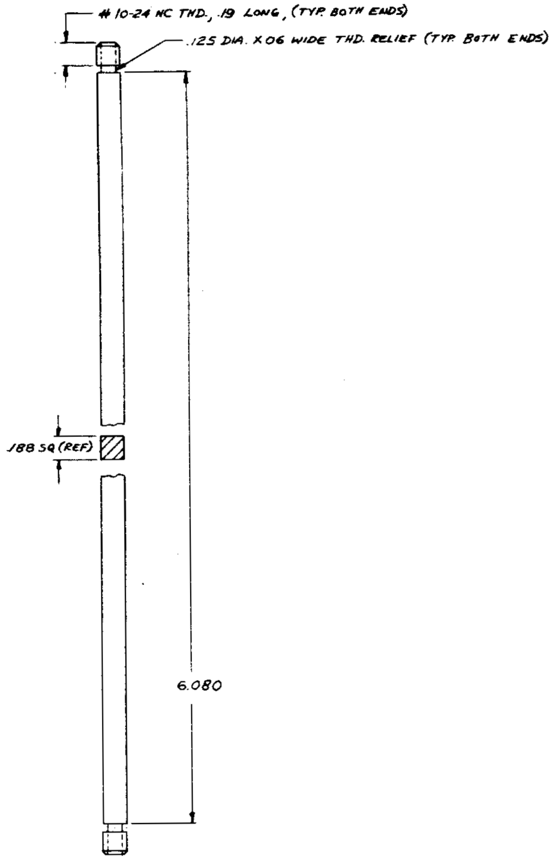
MATERIAL - 3/8" O.D. X .035 WALL TYPE 347 CRES TUBING

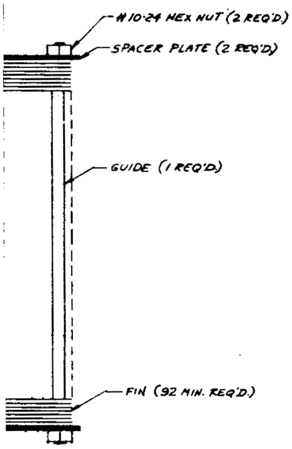
Fig. 16 - Regenerator
Rectangular



SPACER PLATE

MAT'L. - .063 THICK 347 CRES SHEET



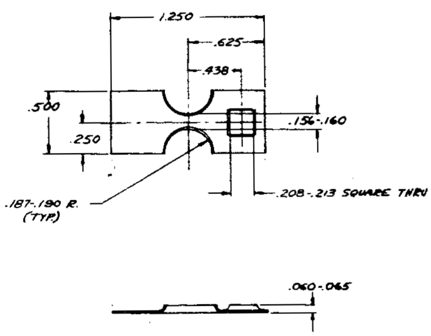


N ASSEMBLY

) EA. REQ'D.

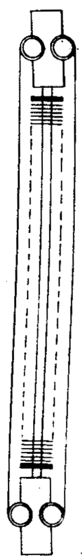
GUIDE

MAT'L. - 3/16 SQUARE TYPE 303 CRES STEEL



FIN

MAT'L. - .0135 COPPER SHEET



Requests for quotation were submitted to the following companies:

1. Cosmodyne Corporation
2. Harrison Radiation
3. Houston Fearless Corporation
4. Northwest Industries Incorporated
5. Precision Sheet Metal Incorporated

Note: The reference curve attached to the specification is similar to Fig. 14 of Progress Report PR 91565-430-3 except that all curves other than the hydrogen and exhaust gas lines have been deleted, and the hot end approach temperature has been raised from 50°R to 100°R.

B. Design Study

The regenerator design study resulted in a design layout consisting of two parallel and identical engine exhaust gas to hydrogen exchangers (Fig. 11) and one hydrogen to oxygen exchanger (Fig. 12). The physical arrangement of the three exchangers is shown schematically in Fig. 13.

The decision to use a separate hydrogen to oxygen heat exchanger, rather than to have one exchanger in which both the hydrogen and oxygen were heated by the exhaust, was made for the following reasons:

1. Simpler (and thus faster and cheaper) fabrication.
2. Safety. The possibility of oxygen leaking into the hot exhaust gas flow area and causing a serious fire or explosion is eliminated.

3. Simplified testing and data evaluation.

By inspection of Figs. 11, 12, and 13 it can be seen that the complete regenerator could be housed in a package not much larger than one exhaust gas to hydrogen exchanger (Fig. 11) by coiling two parallel concentric exhaust gas to hydrogen exchangers around the hydrogen to oxygen exchanger. Considering that the test design is conservative (assuming mean dry exhaust gas properties and steady state film coefficients), a flight design based upon the same fabrication principles should be considerably smaller than one of the engine exhaust gas to hydrogen exchangers shown in Fig. 11.

The exhaust gas to hydrogen exchanger shown in Fig. 11 consists essentially of a helical coil of 3/8" O.D., 0.035" wall, stainless steel tubing to which rectangular copper fins are attached (by silver or nickel furnace braze) with a spacing of 15 fins per inch over a total of 22 3/4 feet of the tube length. The engine exhaust gas is constrained to flow through the finned annulus by the inner and outer stainless steel shells. Thermocouples will be used to measure wall temperature needed to evaluate exhaust gas heat transfer coefficients. These thermocouples are attached to the down stream side of the outer tube wall at each end, in the middle, and near the last 10% (where condensation will occur) of the exchanger length. The exchanger is fabricated in the following sequence of operations:

1. The coil is wound.
2. Fins are attached.
3. The inner shell is inserted. Stainless steel fins will be provided at suitable location to prevent damaging of the copper fins during assembly.

4. The fins are clipped and holes are drilled through the inner shell where thermocouples are to be located.
5. Thermocouples are inserted and welded in place.
6. The outer shell is attached and the flange is welded. The thermocouples may be replaced by removing the outer shell.

A compilation of calculations and results upon which the design is based is given in Appendix B.

During the design study, other configurations based on the same tube size and fin design were considered, Figs. 14, 15, and 16 show preliminary design studies of two flat configurations and one square, helical configuration. The basic advantage of a flat configuration is that fins can be assembled in stacks on rods, and then held to (or held by) the tubing for welding. However, the complex tube bending and housings required for these three configurations, coupled with the need to hold close tolerance between the fins and tubing to assure good weld contact, makes these three designs compare unfavorably with a circular helical design for the purposes of this program. Special fins made (at no cost to Vickers) by the Gasket Manufacturing Company, Inc., of the washer, stock, and ribbon types were used in the evaluation of the different design approaches.

A more compact design can be achieved by the use of specially formed tubing and by manifolding or internally welding tubes. Such a design should be considered for a flight system. Due to expense, time, and safety considerations, one of the design objectives for the test generator was that it should have no internal tube connections. This design objective, in conjunction with a reasonable size limitation, eliminated the finless approach.

The hydrogen to oxygen heat exchanger shown in Fig. 12 is an arbitrary design consisting of brazed, concentric hydrogen and oxygen coils. The calculations shown in Appendix C indicate that this exchanger is more than sufficient to bring the exit hydrogen and oxygen temperatures within a few degrees of each other. Although heating oxygen to the hydrogen temperature does nothing to affect the thermal performance of the system, it may aid in metering control of the propellants.

C. Test Set-Up

The regenerator test circuit is being designed. The exhaust gas flow will be supplied by a water cooled low pressure combustor. The possibilities of using steady state, or pulsating exhaust flow is being considered.

APPENDIX A

REGENERATOR
SPECIFICATION

1.0 GENERAL: The regenerator is to be used during the breadboard component development test phase of the hydrogen-oxygen engine, auxiliary electric power supply development program,which Vickers is performing under NASA Contract NAS 3-2550.

2.0 SCOPE: This specification describes the requirements for a test-type regenerator. The regenerator is to be used to preheat engine propellants (hydrogen and oxygen) by transferring heat to them from the engine exhaust (H_2 and H_2O). Early delivery and price are of primary importance. The use of standard parts or the modification of standard parts is desirable.

3.0 TECHNICAL INFORMATION AND REQUIREMENTS

3.1 Engine Exhaust Gas Data

Inlet temperature will be = 1540°R

Inlet pressure will be = 2.0 psia

Ambient pressure will be = 0 psi

Note: The regenerator shall be designed for the specified thermal performance with 2 psia inlet pressure and with the exit discharging to a space vacuum; however, initial tests will be performed with 14.7 psia back pressure.

The pressure drop to inlet pressure ratio:

$$\frac{\Delta P}{P} = 20\% \text{ max.} \quad \text{(0.4 psi with 2.0 psia inlet)}$$

Weight rate of flow = 42.2 lb. per hr.

3.2 <u>Hydrogen Data</u>

Inlet temperature will be = 250°R

Exit temperature shall be = 1440°R (Ref.)

Inlet pressure will be = 1200 psi

The pressure drop to inlet pressure ratio:

$$\frac{\Delta P}{P} = 5\% \text{ max.}$$

The total cross-sectional flow area shall be not less than <u>0.146 sq. in.</u>, i.e., 5/8" OD or larger if a single tube is used, or 3/8" OD or larger if two parallel tubes are used. (The area limitation is dictated by the engine hydrogen valve area requirement)

The weight rate of flow will be = 14.1 lbs per hour.

3.3 <u>Oxygen Data</u>

Inlet temperature will be = 250°R

Outlet temperature shall be = 1250°R min.

Inlet pressure will be = 1500 psia

The pressure drop to inlet pressure ratio:

$$\frac{\Delta P}{P} = 5\% \text{ max.}$$

Minimum cross-sectional flow area = 0.01 sq. in.

Oxygen to hydrogen weight ratio = 2

3.4 Heat Transfer Requirements

3.4.1 Hydrogen

The hydrogen part of the regenerator shall be designed assuming that all heat transfer takes place between the engine exhaust gas and the hydrogen (no oxygen flowing).

Hot end approach temperature:

δt = 100°R max.

Heat transfer rate approximately

= 58,400 BTU per hour (Ref.)

Note: The above heat transfer rate is the result of Vickers' preliminary calculations and is submitted as a reference only.

Mean dry gas properties shall be assumed for the engine exhaust gas when calculating the heat transfer between the engine exhaust gas and the hydrogen, i.e., neglect knee in the exhaust gas curve as shown on the attached reference plot.

3.4.2 Oxygen

The regenerator shall be designed to raise the oxygen temperature from 250°R to 1250°R or more while the hydrogen is also flowing. The hot end approach temperature St of the ~~hydrogen~~ (see 3.4) will increase when oxygen is flowing.

4.0 INSTRUMENTATION

The regenerator shall have a sufficient number of thermocouples to obtain the average engine exhaust gas wall temperature (the exhaust gas side of the hydrogen tube) at the following stations:

a - the hot end.

b - the cold end.

c - the middle.

d - 10% of the length from the cold end.

The thermocouples should be located so that they are not affected by the exhaust gas temperature.

Parts shall be provided for thermocouple lead wires, and the regenerator shall be designed so that thermocouples can be replaced.

5.0 VENDOR TESTING AND DESIGN VERIFICATION

The vendor is not required to perform tests with hydrogen, oxygen and exhaust gas, or tests to demonstrate performance. The vendor need perform only those tests he considers necessary to guarantee the structural integrity of the regenerator under its specified operating conditions. Thermal performance must be demonstrated to Vickers' satisfaction by regenerator design layout and analysis.

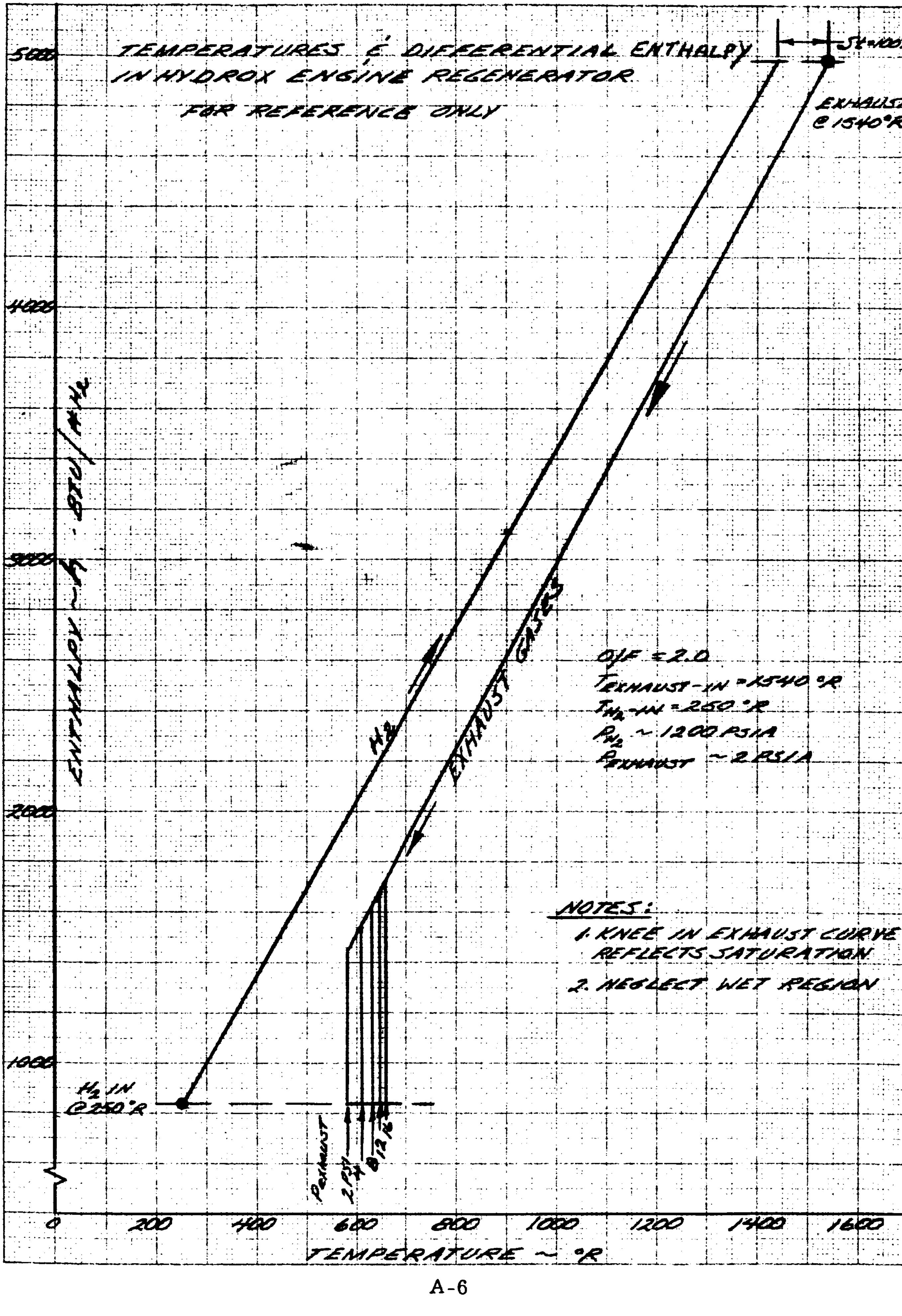
6.0 SAFETY

The regenerator shall be designed with a safety factor of six (6) at peak operating temperatures and pressures.

The regenerator should be designed so that if a leak should occur in the oxygen line, oxygen will leak to ambient and will not leak to the hydrogen or engine exhaust (which contains hydrogen) flow areas.

7.0 CONFIGURATION

The detail configuration is left to the vendor. Since the regenerator is to be used for extensive bench testing, suitable mounting brackets shall be provided. Size and weight are not critical; however, the regenerator should reflect the performance that would be anticipated in a flight weight unit.



TEMPERATURES & DIFFERENTIAL ENTHALPY
IN HYDROX ENGINE REGENERATOR
FOR REFERENCE ONLY
ΔT = 100°R
EXHAUST
@ 1540°R
H2
EXHAUST GASES
O/F = 2.0
T EXHAUST-IN = 1540 °R
T H2-IN = 250 °R
P H2 ~ 1200 PSIA
P EXHAUST ~ 2 PSIA
NOTES:
1. KNEE IN EXHAUST CURVE REFLECTS SATURATION
2. NEGLECT WET REGION
H2 IN
@ 250°R
P EXHAUST
2 PSI
4
8
12
16
5000
4000
3000
2000
1000
ENTHALPY ~ H - BTU/#H2
0
200
400
600
800
1000
1200
1400
1600
TEMPERATURE ~ °R

APPENDIX B

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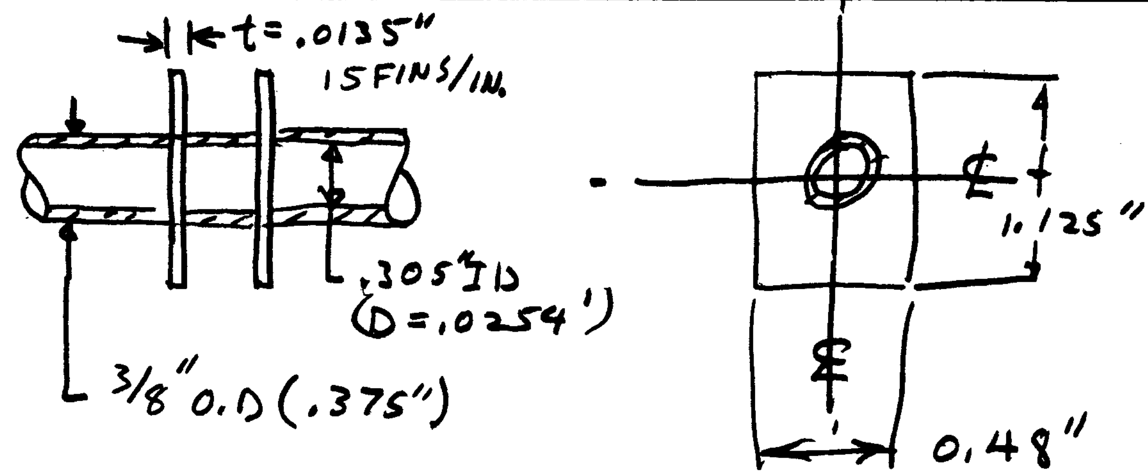
PREPARED BY A. FINK	**VICKERS INCORPORATED** ENGINEERING CALCULATION FORM	PAGE 1 OF 6
CHECKED BY & COPIED W. D. MORATH	SUBJECT EXHAUST GAS TO H_2 HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE



TWO || 22.75' LENGTHS OF 3/8" TUBING REQUIRER. ∴ ACTIVE LENGTH = 45.5'

INSIDE SURFACE AREA (H_2 AREA)

$$A_{H_2} = .08 \frac{ft^2}{ft} \times 45.5 = 3.64\ Ft^2$$

OUTSIDE SURFACE AREA:

$$A_{EX} = 1.06\ ft^2/ft \times 45.5 = 48.2$$

INSIDE CROSS SECTIONAL AREA

$$S_{H_2} = .508 \times 10^{-3}\ FT^2$$

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PREPARED BY A. FINK	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 2 OF 6
CHECKED BY & COPIED WDM	SUBJECT EXHAUST GAS TO H_2 HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

GIVEN CONDITIONS:

$\overset{\circ}{\omega}_{H_2} = 14.1$ LB/HR.

$\overset{\circ}{\omega}_{EX.} = 42.3$ LB/HR.

INLET EXHAUST TEMP:

$T_{EX\ IN} = 1540°R$

INLET H_2 TEMP.

$T_{H_2 IN} = 250°R$

REQ'D. H_2 OUTLET TEMP

$T_{H_2 OUT} = 1440°R$

$$q_E = C_{P_{H_2}} \dot{\omega}_{H_2} \Delta T_{H_2} = 3.48 \times 14.1 \times 1190$$

$$= 58{,}400 \text{ B/HR.}$$

HYDROGEN SIDE:

$$G = \frac{\overset{\circ}{\omega}_{H_2}}{S_{H_2}} = \frac{14.1 \times 10^3}{.508} = 27.8 \times 10^3$$

$$\left(\frac{C_P \mu}{K}\right)^{.4} = \left(\frac{3.48 \times .029}{.0960}\right)^{.4} = (1.05)^{.4} = 1.02$$

$$\left(\frac{DG}{\mu}\right)^{.8} = \left(\frac{.0254 \times 27.8 \times 10^3}{.029}\right)^{.8} = (24{,}400)^{.8}$$

$$= 3{,}250$$

PREPARED BY A. FINK	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 3 OF 6
CHECKED BY & COPIED W D M.	SUBJECT EXHAUST GAS TO H_2 HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

FILM COEFF.

$$h_{f H_2} = .023 \frac{K}{D} \left(\frac{DG}{\mu}\right)^{.8} \left(\frac{c_p \mu}{K}\right)^{.4}$$

$$= .023 \times \frac{.0966}{.0254} \times 1.02 \times 3{,}250 = 296$$

$$q_E = h_f A_{H_2} \Theta_{film}$$

$$\therefore \Theta_{film} = \frac{58{,}400}{3.64 \times 296} = 57 \,°\Delta R$$

EXHAUST GAS SIDE:

BLOCKAGE FACTOR $\left(\frac{\text{FLOW AREA}}{\text{BLOCKED AREA}}\right)$

$$F_2 = (1.-.333)(1 - t' \times \frac{\text{FINS}}{\text{IN}})$$

$$= .667 \times (1 - .0135 \times 15) = 0.533$$

WETTED AREA / VOLUME FACTOR

$$F_s = \frac{1.06 \text{ ft}^2}{\text{ft}} \times \frac{1728 \text{ ft}}{1.125 \times .45 \times 12 \text{ ft}^3}$$

$$= 300$$

$$G_o = \frac{\dot{W}_{EX}}{15 F_s} = \frac{423}{15 \times .533}(144) = 762 \,\#/\text{hr ft}^2$$

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PREPARED BY A. FINK	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 4 OF 6
CHECKED BY ∮ COPIED WRM	SUBJECT EXHAUST GAS TO H_2 HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

FILM COEFF.:

$$*\ h_{f_{EX}} = \frac{C_{p\,MEAN}}{2.21}(G_0)^{.22}\left(\frac{\mu F_1}{F_2}\right)^{.78}$$

$$= \frac{1.22}{2.21}(762)^{.22}\left(\frac{.028 \times 300}{.533}\right)^{.78}$$

$$= \frac{1.22}{2.21} \times 4.31 \times 8.6 = 20\ B/hr\text{-}ft^2\,^\circ R$$

FOR MINIMUM FIN LENGTH:

$$L = 0.375'' = .0312'$$

HALF SECTIONAL AREA:

$$a_L = \frac{1}{2}\left(\frac{.0135}{12}\right)(.0312)$$

$$= 0.1755(10^{-4})\ Ft^2$$

$$k_{COPPER} = 196\ B/hr\ ft^2\ ^\circ F/FT$$

$$\frac{h_f L}{k a_L} = \frac{19(.0312) \times 10^4}{196(.1755)} = 173$$

* FROM page 6, "ANALYTICAL METHOD FOR COMPACT REGENERATORS" BY H. WOOD

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PREPARED BY A. FINK	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 5 OF 6
CHECKED BY § CHECKED W. D. M	SUBJECT EXHAUST GAS TO H_2 HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

$$h = L\sqrt{\frac{h_f L}{k A_L}} = .0312\sqrt{173} = 0.41$$

*

FIN EFFECTIVENESS

$$\eta_f = 0.94$$

ALLOW AN ADDITIONAL 10% FOR "IN LINE" FLOW BLOCKAGE

$$\text{LET } \eta_{fin} = 0.9(0.94) \cong 0.85$$

TEMPERATURE DROP ACROSS FILM

$$\theta_{f\,EX} = \frac{q_x}{A_{EX}\,\eta_f\,h_{f\,EX}} =$$

$$\frac{58,400}{48.2 \times .85 \times 20} = 70.5°\Delta R$$

Ln. MEAN TEMPERATURE DROP ACROSS EXCHANGER (NEGLECT METAL)

$$(\Delta T)_M = 57 + 70.5 = 126.5°\Delta R$$

OK ✓

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* fig 11-12 Page 236 JACOB

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ASSUME MEAN DRY GAS EXHAUST CONDITIONS AND CHECK $(\Delta T)_M$.

$$\Delta T_{EX} = \frac{q_c}{C_{P_{EX}} \dot{\omega}_{EX}} = \frac{58{,}400}{1.22 \times 423}$$

$$= 1{,}130\ ^\circ R \quad \therefore \quad T_{EX_{EXIT}} = 410\ ^\circ R$$

COLD END $\delta T_c = 410 - 250 = 190\ ^\circ R$

HOT END $\delta T_H = 1540 - 1440 = 100\ ^\circ R$

$$(\Delta T)_M = \frac{190 - 100}{\ln \frac{190}{100}} = \frac{90}{.642} = 140\ ^\circ R$$

OK

EXHAUST GAS PRESSURE DROP

$$* \quad \Delta P = 4 \times 10^{-9} L \frac{1}{\rho_A} (\mu)^{.95} G^{1.05} \left(\frac{F_1}{F_2}\right)^{1.95}$$

Where ρ_A = mean density

$$\Delta P = 4 \cdot 10^{-9} \frac{18.45}{12} \times \frac{1}{.0011} (.0286)^{.95} (762)^{1.05} (675)^{1.95}$$

(18.45: assumed)

$$= 65\ \#/FT^2 = 0.45\ PSI$$

* Page 7 "ANALYTICAL METHOD FOR COMPACT REGENERATORS" REPORT # NSW 130 BY H. WOOD

APPENDIX C

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PREPARED BY	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 1 OF 4
MORATH	SUBJECT	PROGRAM & PROJECT NO.
CHECKED BY	H_2 TO O_2 TEST HEAT EXCHANGER	91565-439
		DATE

ASSUMING: (1) A H_2 COIL OF 7 TURNS OF 5/8" O.D., .049" WALL TUBING ON A 3" MEAN DIA.

(2) A O_2 COIL OF 12 TURNS OF 3/8" O.D. .035" WALL TUBING ON A 4" MEAN DIA.

(3) PERFECT CONDUCTION OF THE METAL

(4) FLUIDS FLOWING IN THE SAME DIRECTION

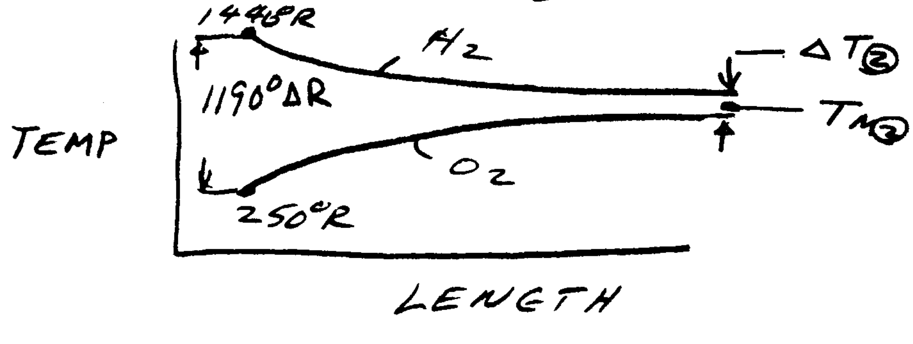
GIVEN: AN INLET H_2 TEMPERATURE $T_{H_2 1} = 1440\,°R$

& AN INLET O_2 TEMPERATURE OF $T_{O_2 1} = 250\,°R$

RESULTING IN AN INLET $\Delta T_1 = 1190\,°\Delta R$

& $\dot{\omega}_{H_2} = 14.1$ LB/HR & $\dot{\omega}_{O_2} = 28.2$ LB/HR

FIND: THE APPROX. OUTLET ΔT_2 & THE APPROX MAGNITUDE OF THE OUTLET TEMP. T_{M2}



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PREPARED BY	VICKERS INCORPORATED	PAGE 2 OF 4
MORATH	ENGINEERING CALCULATION FORM	PROGRAM & PROJECT NO. 91565-439
CHECKED BY	SUBJECT: H_2 TO O_2 TEST HEAT EXCHANGER	DATE

<u>HYDROGEN COIL:</u>

<u>INSIDE DIA:</u> $D_{iH_2} = 0.625 - 2(.049) = .527'' = .0439'$

<u>CROSS SECTIONAL AREA:</u>

$$S_{H_2} = \frac{0.218128}{144} = 1.514 \times 10^{-3}\ ft^2$$

<u>TUBE LENGTH:</u> $l_{H_2} = \frac{3'' \pi 7}{12} = 5.5\ FT$

<u>INSIDE SURFACE AREA:</u>

$$A_{iH_2} = l_{H_2} C_{H_2} = 5.5 \times 0.1475 = 0.81'$$

<u>OXYGEN:</u>

<u>INSIDE DIA:</u> $D_{iO_2} = 0.375 - 2(.035) = .305'' = 0.0254'$

<u>CROSS SECTIONAL AREA:</u>

$$S_{O_2} = \frac{.073062}{144} = 0.508 \times 10^{-3}\ FT^2$$

<u>TUBE LENGTH:</u> $l_{O_2} = \frac{4'' \pi 12}{12} = 12.58'$

<u>INSIDE SURFACE AREA:</u>

$$A_{O_2} = l_{O_2} C_{O_2} = 12.58 \times .08 = 1.03\ FT^2$$

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PREPARED BY MORATH	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 3 OF 4
CHECKED BY	SUBJECT H_2 TO O_2 TEST HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

	H_2	O_2
ASSUME: VISCOSITY, μ =	0.029	0.0653
CONDUCTION k =	.0966	.0254
SPECIFIC HEAT C_p =	3.42	0.217
$G = \dot{w}/s$ =	$\frac{14.1}{1.514} \times 10^3 = 9.3 \times 10^3$	$\frac{28.2 \times 10^3}{.508} = 55.3 \times 10^3$
$\left(\frac{C_p \mu}{k}\right)^{0.4}$ =	$\left(\frac{3.42 \times .029}{.0966}\right)^{.4} = 1.104$	$\left(\frac{.217 \times .0653}{.0142}\right)^{.4} = 1$
$\left(\frac{DG}{\mu}\right)^{.8}$ =	$\left(\frac{.0439 \times 9.3 \times 10^3}{.029}\right)^{.8}$? $= 2,100$	$\left(\frac{.0254 \times 55.3 \times 10^3}{.0653}\right)^{.8}$ $= 2,800$
FILM COEFF.:		
h =	$.023 \frac{k}{D} \left(\frac{DG}{\mu}\right)^{.8} \left(\frac{C_p \mu}{k}\right)^{.4}$ $= \frac{.023 \times .0966 (1.04)(2,100)}{.0439}$ $h_{H_2} = 118$	$\frac{.023 \times .0142 (1)(2,800)}{.0254}$ $h_{O_2} = 36$

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PREPARED BY MORATH	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 4 OF 4
CHECKED BY	SUBJECT H_2 TO O_2 TEST HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

$$\frac{1}{UA} = \frac{1}{h_{H_2}} + \frac{1}{h_{O_2}A_{O_2}} = \frac{1}{118 \times .81} + \frac{1}{36 \times 1.03}$$

$$= .03546$$

$$UA = 28.2$$

FIND $T_{M②}$: AT ∞ LENGTH

$$T_{M③} = T_{H_2③} = T_{O_2②}$$

$$q = C_{P_{H_2}} \dot{\omega}_{H_2} \Delta T_{H_2} = C_{P_{O_2}} \dot{\omega}_{O_2} \Delta T_{H_2}$$

$$\Delta T_{O_2} = \frac{3.42 \times 14.1}{.217 \times 28.2} \Delta T_{H_2} = 7.93 \, \Delta t_{H_2}$$

$$250 + \Delta T_{O_2} = 1440 - \Delta T_{H_2}$$

$$\Delta T_{H_2} = \frac{1190}{8.93} \approx 133 \, ^\circ\Delta R$$

$$T_{M②} \approx 1440 - 133 = 1307\,^\circ R$$

APPROX HEAT TRANSFER RATE:

$$q = (C_P \dot{\omega} \Delta t)_{H_2} = 3.42 \times 14.1 \times 133$$

$$= 6400 \, B/HR$$

$$q = UA (\Delta t)_M \therefore (\Delta t)_M = \frac{6400}{28.2} = 227\,^\circ\Delta R$$

TRY $\Delta T_② = \boxed{6\,^\circ\Delta R}$; OK

$$(\Delta t)_M = \frac{1190 - 6}{\ln \frac{1190}{20}} = \frac{1184}{\ln \frac{1190}{6}} = \frac{1184}{\ln 198.5}$$

$$= \frac{1184}{5.28} = 224\,^\circ\Delta R$$